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BROADENING THE FIELD OF THE MARINE STEAM TURBINE:

THE PROBLEM, AND ITS SOLUTION

Mr. Geo. F. Baker

Compts Richard Goodman

BROADENING THE FIELD OF THE MARINE STEAM TURBINE: THE PROBLEM, AND ITS SOLUTION

THE MELVILLE AND MACALPINE REDUCTION-GEAR

Printed for
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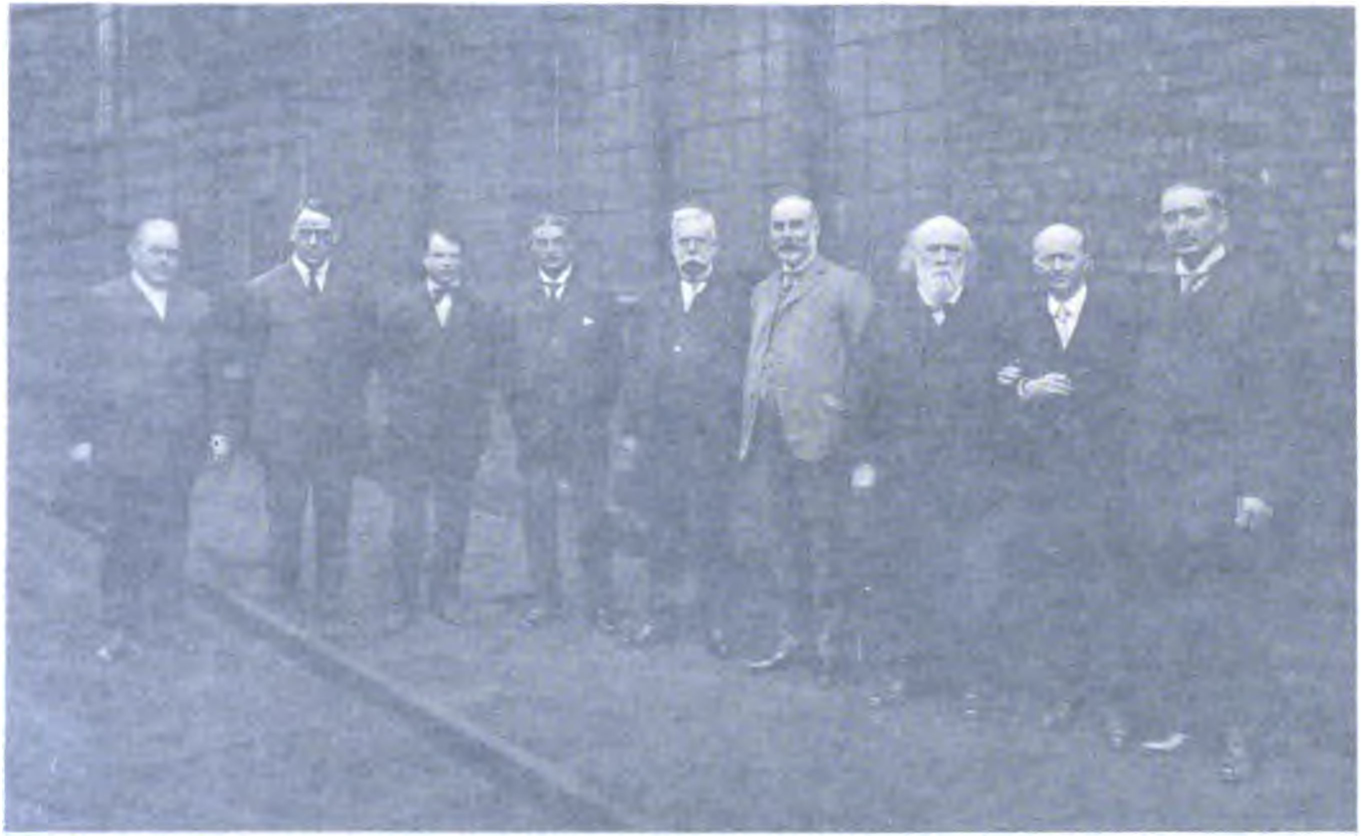

PARTY WITNESSING FIRST FULL LOAD TESTS OF MELVILLE AND MACALPINE GEAR.

From left to right: [illegible] W. M. McFarland, [illegible] Westinghouse Electric & Mfg. Co.; H. T. Herr, General Manager The Westinghouse Machine [illegible] N. [illegible] W. [illegible] Manager [illegible] New York Shipbuilding Co.; George Westinghouse; [illegible] N. [illegible] George W. Melville, Ex-Engineer-in-Chief U. S. Navy; John H. Macalpine, M. I. N. A., [illegible] of Engineer [illegible]

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Broadening the Field of the Marine Steam Turbine

A Note Added Since Publication

The owners and builders of turbine propelled ships have generally been secretive in regard to steam consumption, but the lack of public information may have been due to an uncertainty as to the steam economy tests made under working conditions. The difficulties involved in measuring such large powers, especially when it is remembered that the only indication is the twist in a ten foot length of shaft, are so considerable that it is reasonable to suppose that engineers would not commit themselves until, by repeated experiments, they had acquired full confidence in the reliability of this unusual method of measurement to which they are limited.

At the time when the introductory to this publication was written, no definite information as to the steam consumption of any of the important marine installations was available. The figure of 14½ pounds per shaft horse-power was assumed for the reason that the latest tenders made to the United States Government for battleships to be propelled by turbines made according to plans supplied by the owners of Mr. Parsons' U. S. patents, specified a steam consumption not exceeding 14.7 pounds per shaft horse-power, and it naturally would not be expected that this water rate could be greatly bettered.

However, we have since been favored with specific information in the able and comprehensive presidential address on the "Propelling Machinery of Warships," delivered before the Junior Institution of Engineers on November 16, 1909, by Engineer Vice-Admiral H. J. Oram, C. B., R. N., Engineer-in-Chief of the Fleet.

Admiral Oram's sound engineering judgment, coupled with unusual opportunities for following the progress of the marine steam turbine in the Royal Navy, are such that I fully accept his statements, and record my acknowledgment of the value of his address by making a brief quotation from that section in which he deals with the efficiency of the turbine, and which includes his graceful and well deserved tribute to the genius and persevering energy of the Honorable Charles Algernon Parsons.

Although the reciprocating engine in the Navy, in its final state of development was of considerable excellence, it had to make way for the marine steam-turbine, which is due to the genius, enterprise, and patient research of one of our Past-Presidents. All recent warships in the Navy are fitted with turbine engines; and, as illustrating the magnitude of the change recently made, it may be mentioned that there are now either actually constructed, or on order, practically 2,000,000 S. H.-P. of Parsons turbines for the British Navy alone, and I am sure we should all wish to congratulate Mr. Parsons thereon. Now that everything is progressing smoothly as regards turbine machinery, and as ship after ship carries out successful trials without incident or failure, we are apt to forget the years of strenuous work and the magnitude of the difficulties faced and surmounted by him in the earlier stages.

The great success and freedom from accident which has attended this great revolution in propelling machinery and the application of new principles on such an extensive scale, is remarkable, and it is due to Mr. Parsons to acknowledge the skill in design which has enabled these good results to be achieved and the estimates of performance of marine turbines to be amply borne out by actual results.

Besides the gain in economy at full power resulting in a diminution in numbers of boilers, and the comparative freedom of the turbine from running defects, the clean and dry engine-room platforms, absence of noise and bustle, etc., are features which are noticeable by everyone familiar with corresponding conditions of reciprocating engines.

The steam-engine indicator also, invaluable for reciprocating engines, is of little or no service with turbines. Its place as regards the estimation of power has been taken by the torsion-meter, and it is astonishing with what rapidity this new instrument has been developed to meet the requirements of steam-turbines.

Trials of the turbine cruiser Amethyst furnished a good comparison between reciprocating machinery and turbines in the small cruiser class, but the first turbine machinery fitted in the Navy of large size was in the Dreadnought, and it is well known how rapidly this machinery and ship were constructed and what excellent results were obtained.

When it was decided in the Dreadnought to supersede the ordinary reciprocating engine by the Parsons marine steam-turbine, careful analyses of the results obtained in previous vessels indicated that an increased economy would be realized over the average type of reciprocating engine of that period, to an extent which would enable the S. H.-P. of the turbine to be the same as the indicated horse-power of the reciprocating engine, while using about 15 per cent. less steam.

The machinery of the Dreadnought was designed on these lines—*i. e.*, a reduction of about 15 per cent. was made in the usual boiler proportions, and the trial results confirmed the action taken, the consumption of steam at full power for turbines only being 13.48 lb. per S. H.-P. per hour, as carefully ascertained from large measuring-tanks. The initial pressure at the turbines was 164 lbs. per square inch by gauge.

The turbines of subsequent ships have been gradually improved with experience, and although the general construction remains the same, clearances and other adjustments required are by experience better known, and improved fittings and measuring appliances enable the dummy clearances, on which steam leakage very much depends, to be reduced, so that subsequent results have been even more satisfactory than those of the Dreadnought. The average steam consumption of the three battleships succeeding this ship was 13.01 lb. per S. H.-P. per hour for turbines only, with an average gauge pressure of 147 lb. per square inch on the high-pressure turbine.

The turbine engines of the Indomitable class of 41,000 S. H.-P. were much greater in size and of slower revolutions than those of the battleships, and the effect of this on the efficiency of the turbines could not be quite foreseen, so rather a conservative estimate was made of the probable steam economy; and, though the boilers were made a little smaller proportionately (about 4 per cent.) than in the Dreadnought, trials of these ships showed that the steam consumption of the turbines was lower than expected. The results were superior to anything that could have been predicted with confidence at the time, as the average consumption of steam at full power for turbines only was 12.03 lb. per S. H.-P., with the average gauge pressure 123 lb. per square inch at the high-pressure turbine.

GEO. WESTINGHOUSE.

Pittsburg
January, 1910

INTRODUCTORY

ABOUT six years ago, Rear-Admiral George W. Melville, ex-Engineer-in-Chief, U. S. Navy, and Mr. John H. Macalpine, Consulting Engineers, undertook at my request a thorough investigation of the then existing status of the Steam Turbine as applied to the propulsion of ships, and the probabilities of its becoming the ultimate successor of the highly developed types of reciprocating engines commonly used for that purpose.

The data gathered in the course of the investigation and the conclusions drawn therefrom, were embodied in an exhaustive and interesting report which was delivered to me by its authors in May, 1904. Because of the universally recognized high standing of Admiral Melville and his associate as authorities in matters pertaining to marine architecture and engineering, and because the report was such a complete, temperate and logical exposition of the Marine steam turbine situation at that time, I considered that the information collected was of more than passing personal interest, and I had a limited number of copies of the report printed for private circulation, in order that others might benefit by a plain dispassionate statement of fact, stripped of the fanciful embellishments with which a novel departure from established lines of practice is apt to be adorned unconsciously by over-enthusiastic advocates.

The report of May, 1904, was not of a very encouraging nature, but in the few years that have intervened, there have been developments which effect a radical change in the situation existing at that time. This original report has been incorporated as a supplement to the accompanying description of certain interesting and important work that has been carried out by The Westinghouse Machine Company at East Pittsburg, Pa., under the direction of Messrs. Melville and Macalpine, and by comparison, it gives one a vivid impression of the rapidity with which history is made in the engineering art.

The most significant statement in the report of May, 1904, is to be found in the conclusion. It reads as follows:—

"*If one could devise a means of reconciling, in a practical manner, the necessary high speed of revolution of the turbine with the comparatively low rate of revolution required by an efficient propeller, the problem would be solved, and the turbine would practically wipe out the reciprocating engine for the propulsion of ships. The solution of this problem would be a stroke of great genius.*"

There could be no greater tribute to the far-seeing and well-balanced judgment of the authors of this report, than the fact that nearly five years later this identical sentiment was publicly voiced not only by Mr. James Denny of Denny & Brothers, Dumbarton, Scotland, the oldest and most experienced builders of turbine propelled vessels in the world, and the most consistent advocates of the system, but also by the Honorable Charles Algernon Parsons himself, by whose brilliant creative imagination, the basic idea of the modern marine turbine was conceived, and by whose courageous initiative it was made an accomplished fact.

The parallelism of thought is so unusually striking that I quote the utterances of Mr. Parsons and Mr. Denny verbatim, as reported. Mr. Parsons in concluding his James Watt Anniversary lecture delivered at Greenock, Scotland on January 15th of the present year said:—

"*We might naturally speculate as to the future, and inquire if there is a possibility of the turbine being constructed to run more slowly, and without loss of economy or whether the propeller can be modified to allow of higher speeds of revolution. Or, again, may the solution be found in reverting to some description of gearing—not the primitive wooden spur gearing of half a century ago, but to steel gearing cut by modern machinery with extreme accuracy and running in an oil bath, helical tooth gearing or chain gearing, or again, some form of electrical or hydraulic gearing? These are questions which are receiving attention in some quarters at the present time, and if a satisfactory solution can be found, then the field of the turbine at sea will be further extended.*"

Mr. Denny in his Presidential Address to the Institution of Marine Engineers in Britain on October 5, 1908, said:—

"*It has frequently been suggested that if some inspired engineer would evolve a system of gearing that would be lasting and reliable, not too noisy, and would not absorb in friction more than say, 10 per cent. of the power, turbine engines would be capable of application to any speed of vessel, and to any size of propeller; you would then have a high speed turbine and a low speed propeller, which is the ideal condition for marine propulsion.*"

The problem first recognized by Messrs. Melville and Macalpine, and the importance of which has since been admitted by Mr. Denny and Mr. Parsons, has been solved. It has been solved by Messrs. Melville and Macalpine themselves, so that by one of the amusing freaks of fate, the compliment to the then unknown solver, implied in the last sentence of the paragraph quoted from their report of May, 1904, comes back to rest upon its unsuspecting makers.

The desired end has been accomplished by means of a reduction-gear, which makes possible any reasonable speed ratio between the turbine shaft and the propeller shaft. It was no easy task to design a system of gearing that will operate quietly and without destructive wear at the speeds common to steam turbines of the highest efficiency, and at the same time be capable of transmitting thousands of horse-power. The details of the design that has proven itself capable of fulfilling these requirements, are fully set forth in the able and comprehensive article reprinted from ENGINEERING (London, September 17, 1909), which is based on data supplied by Messrs. Melville and Macalpine. However, a few additional words of a non-technical character, generally descriptive of the gear, and indicative of its significance as a factor in marine construction will not be out of place.

The teeth of the gears are helical, that is to say they do not run straight across the face of the wheel parallel to the axis as in the case of ordinary spur gears, but they are cut in the form of a steep spiral, like an exaggerated screw thread. This construction allows the teeth to roll into contact without shock or jar. If there were only a single gear on each shaft this helical form of tooth would cause an objectionable end thrust. As the gears must be very wide to transmit the enormous powers required in marine service two gears, each of half the required width, are placed on each shaft, with the spirals of the teeth running in opposite directions. In this way the end thrust due to the obliquity of the teeth is completely balanced. With a pair of wide-faced gears with straight teeth, it is hardly possible to cut the teeth with such accuracy and to align the shafts so perfectly as to get uniform contact throughout the entire length. Even if it were possible to secure the requisite degree of accuracy at the outset, it could not be permanently maintained on account of the natural wear of the bearings. In general, the conditions are such that a rigidly confined set of gears such as are common for moderate speeds and powers, is altogether inadmissable.

In the design which has proven its sufficiency under severe and exhaustive tests, the smaller gear or pinion is mounted in what the inventors call a "floating frame." The frame which carries the bearings for the pinion is a heavy steel casting supported only at a single point midway between the bearings. This support is flexible so that the frame is free to oscillate in a vertical plane passing through the axis of the pinion, but is held securely against motion in any other direction. Furthermore, the pinion is free to move endwise in its bearings. Any tendency of the teeth to bear harder at one end of the gear than the other would tend to unbalance the respective end thrusts due to the right and left hand spirals of the teeth; but as the pinion cannot present any resistance to unbalanced end thrust, it constantly adjusts itself in the direction of its axis to the position corresponding to equilibrium between the opposing forces. This means that the tooth contact pressures are always automatically equalized.

If there are any minute irregularities in the spacing of the teeth, which would tend to make the contact harder at one point than another in any part of the revolution, this tendency is defeated by the floating frame, the position of which about its central support or fulcrum is controlled solely by the pressures of the teeth of the pinion against the teeth of the large gear. Naturally, the floating frame always yields under the slightest tendency of an unbalanced contact pressure in such a way as to transfer the smallest increment of unbalancing pressure, to another section of the gear that in the absence of the floating frame would be less inclined to take its full share of the stress. In short, the gears are self adjusting to relieve and equalize all abnormal strains, and are consequently independent of the small inaccuracies that are impossible to eliminate in the best commercial manufacturing operations.

The probable efficiency of the gear was naturally the most anxious question, as the operating conditions were so wholly unprecedented that there were no existing data to enable one to even hazard an estimate. The effect of a low efficiency would be more serious than a mere impairment of the economic performance of the installation, as the transmission losses would manifest themselves in heating and destructive wear that would mean hopeless failure for the scheme in its entirety.

In order to definitely settle the question as to the practicability of the gear, it was necessary to devise methods, and to design and construct special appliances for testing it under all of the conditions of load and speed that would probably obtain in actual commercial service. The dynamometer for applying and measuring the loads to which the gear was subjected, will, I trust, be considered sufficiently novel and interesting to warrant the extended description that will be found in another part of this publication.

Considering the important bearing of the question of efficiency on the ultimate success or failure of the gear, it is peculiarly gratifying to have found by repeated trial and careful measurement, that the transmission loss hoped for by Mr. Denny, has been divided by seven. To be exact, the efficiency surpasses the more than satisfactory figure of 98.5 per cent., a result that is without doubt inseparably connected with the flexibility and self-adjusting character of the apparatus.

It is needless to say that the gear is enclosed in a substantial casing, that adequate means are provided for its constant and efficient lubrication, and that the ingeniously designed connections between the gear and the turbine effectually prevent the self-adjusting movements of the pinion from communicating any longitudinal or transverse stresses to the turbine shaft.

Now that the mechanical operation of the gear is no longer a matter for speculation, it is interesting to consider its bearing on the design of turbine installations in ships.

The turbines of the giant Cunarders, Mauretania and Lusitania, are supposed to be capable of developing 70,000 shaft horse-power. Even the comparatively low speed at which these turbines run is too high for maximum propeller efficiency. It is hardly possible that the propeller efficiency exceeds 55 per cent. which means that the actual effective propelling power is only about 38,500 horse-power. At a lower speed of revolution, well within the capabilities of the reduction-gear, a propeller could be made that would have an efficiency of not less than 65 per cent. With this improved efficiency, the shaft horse-power required for the same effective propelling power would be somewhat less than 57,000 a saving of almost 15 per cent. This means that without sacrificing in the smallest degree the remarkable speed of these vessels, the boiler equipment could be reduced about one-seventh, as well as the amount of coal burned on each voyage. This would not only result in a very marked saving in capital investment and operating expenses, but would add many tons to the cargo carrying capacity, and add correspondingly to the earning power.

But this estimate, large as it is, is still too modest. With the turbine and the propeller direct connected so that both revolve at the same speed, not only is it necessary to sacrifice the efficiency of the propeller, but the efficiency of the turbine as well.

For equal efficiencies in any two turbines, the number of rows of blades is, roughly speaking, inversely proportional to the squares of the respective peripheral speeds of the rotating elements. The peripheral speed of the rotating elements in the turbines of the Mauretania and Lusitania, is only one-third of the speed common in large turbines used on land. This would mean that to obtain the efficiencies common to the latter, the former would require approximately nine times as many rows of blades, which would make a machine of prohibitive length. To maintain the same speed of revolution and increase the peripheral speed of the turbines of these vessels to the point common in land practice the rotors would have to be nearly forty feet in diameter, which is manifestly beyond the shadow of possibility.

From the best information obtainable, it is believed that the steam consumption of the turbines of the Mauretania and Lusitania cannot be less than 14.5 pounds per shaft horse-power per hour, while it has been demonstrated beyond question that turbines of similar capacity operating at speeds which the reduction-gear makes possible for marine service, the steam consumption does not exceed 11 pounds per shaft horse-power per hour. This means that the boiler capacity could be further reduced from the first estimate of 60,000 h. p. to about 45,000 h. p., and the overall efficiency of the installation would be sufficiently improved to result in a reduction of over 35 per cent. in the coal consumption. It is unofficially reported that the coal consumption of these vessels is about 4,700 tons per voyage. Reckoning the cost of the coal at $3.25 per ton, the saving in coal alone would be $5,300 per voyage, to say nothing of the smaller cost for wages and sustenance for the lesser number of stokers that would be required. The increased cargo capacity resulting not only from a reduction of over 1600 tons in the coal required to be carried on each voyage, but also from the greatly reduced weight of the equipment, and the space necessary for it, is an asset the value of which it is difficult to over-estimate.

If greater speed would be regarded as more attractive than the possible economies mentioned above, it is easy to see that with the same boiler capacity as is now installed in these ships, the better economy of the high speed turbine would make it practicable to use much more powerful propelling machinery without increasing the amount of coal consumed. The additional power in the turbines, together with the greater propeller efficiency that is possible, would easily give the increased speed that would insure the disembarking of transatlantic passengers on the fifth day with certainty and regularity, instead of only on occasions when all of the conditions are unusually favorable.

The Mauretania and Lusitania have two high pressure and two low pressure turbines, and two reversing turbines working on four shafts. According to the best information obtainable the high pressure turbines have each 128 double rows of blades, and the low pressure turbines 60 double rows each. The total length of the blading exclusive of the relatively small amount in the reversing turbines, is about 115 miles, and the total surface area of the blades is considerably more than three-quarters of an acre, or equal to the sail area of a large ship.

By using the reduction-gear, the same total propulsive power could be installed in three turbines. There is nothing problematical about this statement, as turbines developing the requisite power, and of the same general design as would in connection with the reduction-gear be suited to marine work, have been operating successfully for a long time, and their power and economy are now matters of authentic record. Each turbine would have only 51 double rows of blades, a total in the three turbines of 153, or only 25 in excess of the number of rows in one of the high pressure turbines alone in the present installations on board the Cunard flyers. The total length of the blading in the three high speed turbines would be less than six per cent. of that in the low speed turbines.

Each shaft would be driven by a complete and independent self-contained turbine, and each shaft would have its own reversing turbine so that the entire screw equipment would be available for backing instead of only one-half of it, as is the case in the present arrangement.

However much the new system promises for express steamers in the mercantile marine, it has vastly more important advantages as applied to naval vessels. The express steamer normally runs at its highest speed, and this is the condition for maximum turbine efficiency. This is especially true in the case of turbines connected directly to the propeller shaft, for the reason that as outlined before, the peripheral speeds and number of rows of blades are at best below the requirements of efficient design, and any dropping below the maximum working speed accentuates the bad effect of this deficiency.

On the other hand, in the case of a battleship or a cruiser, maximum speed is only an emergency condition. The normal cruising speed is only about 60 per cent. of the maximum speed, and requires perhaps less than 25 per cent. of the maximum power. It is at the cruising speed that turbine propelled naval vessels have shown to disadvantage as compared with vessels propelled by the best types of reciprocating engines. By reason of the more liberal blading that is possible in a high speed turbine, its economic performance is less sensitive to departures from maximum rotative speed, than is that of the low speed turbine. Furthermore, as the entire expansion of the steam takes place in a single turbine, the total power may be distributed conveniently among three entirely independent units, driving one central and two wing propellers. The central unit alone will suffice for ordinary cruising speeds, and can be operated always at somewhere near its most economical conditions of working.

In naval service, the ability to start the turbines when cold, and quickly bring them to full speed may often be of the very highest importance. With turbines directly connected to the propeller shafts, the lengths and diameters of the rotors and casings are such that in order to prevent serious distortion from unequal heating and expansion, it has been found necessary in practice to bring all of the turbine machinery to the normal working temperature before it may safely be set in motion. I have been informed by those having charge of turbine machinery on a large battleship, that the preliminary warming often requires some hours.

In the double flow turbine which it is proposed to use with the Melville and Macalpine gearing, the smaller dimensions consequent on the higher speeds give a sturdier construction in which the tendency to distortion is reduced to a negligible minimum; and an elastic self-adjusting mounting for the stationary blades, easily removable for examination without unseating the rotor, compensates for any inequality in the expansion of the rotor and the casing, and effectually prevents the stripping of blades even if there should be actual contact between the stationary blades and the body of the rotor, or between the moving blades and the casing. With this construction, steam, even though it carries large quantities of water of condensation with it, may be admitted to cold turbines, and full speed obtained in less than a minute.

The United States Government has lately awarded contracts for two new battleships, to be equipped with steam turbines. These battleships are to have a speed of 20½ knots, which will require in round numbers 28,000 shaft horse-power. With 55 per cent. propeller efficiency the effective propelling power will be about 15,400 horse-power. With the 65 per cent. propeller efficiency that is easily possible with the reduction-gear and a propeller at a lower speed of revolution, this same propelling power would require less than 24,000 horse-power on the shaft. The average steam consumption guaranteed at full power is about 14½ pounds per shaft horse-power. With the better steam economy of the high speed turbine, the boiler capacity required would be reduced fully one third. With the same bunker capacity, the radius of action would be enormously increased, which is an advantage of incalculable value.

If the same boiler equipment as is now proposed were maintained, there would still be a saving in weight of over 250 tons, or approximately one-eighth of the total penalty weight of the machinery in each ship, resulting solely from the substitution of the high speed turbine and reduction-gear for the more cumbersome slow speed direct connected machine. At the same time, by reason of the well known overload capacity of a liberally proportioned turbine, there would be available a surplus power of about 50 per cent., which should make possible an emergency speed of nearly three knots in excess of that called for in the specifications.

Furthermore, the three independent shafts, each with its own self-contained turbines for going ahead and astern, would give the excellent manoeuvering qualities which are admittedly lacking in vessels fitted with the present conventional turbine equipment.

The certainty with which the floating-frame of the Melville and Macalpine reduction-gear operates to maintain an evenness of tooth-pressure and the limitation of the maximum pressure to 450 lbs. per inch of tooth-contact with a load of 6000 h.p., coupled with a large factor of safety, at once remove this invention from an experimental to a completely commercial apparatus.

I regard this invention as epoch-making in its importance. It has been my privilege to supply—the material things which were needed to transform the creature of Messrs. Melville and Macalpine's imagination into an actual thing of iron and steel. The results achieved by the completed machine have fully justified my faith in its ultimate success.

GEO. WESTINGHOUSE.

PITTSBURG, October 15, 1909.

EXPERIMENTAL SPUR-WHEEL REDUCTION-GEAR FOR HIGH-SPEED STEAM-TURBINES

CONSTRUCTED FROM THE DESIGNS OF
REAR-ADMIRAL GEORGE W. MELVILLE
AND MR. JOHN H. MACALPINE

A REPRINT FROM
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LONDON, SEPTEMBER 17, 1909

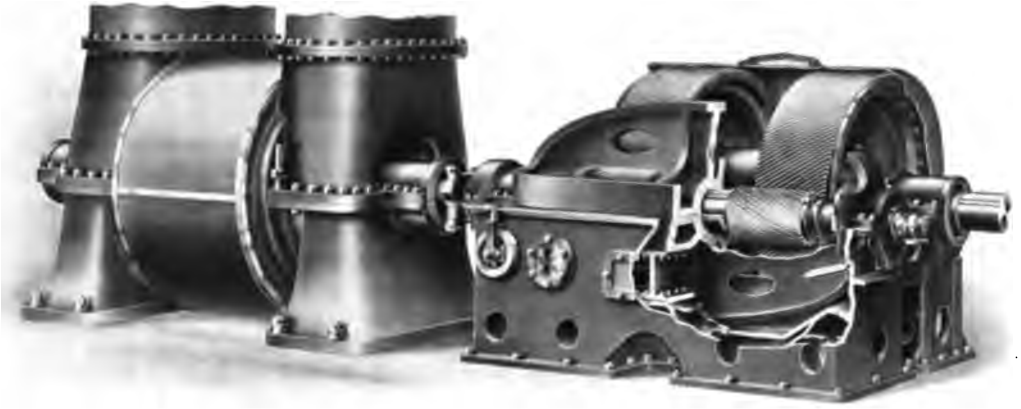


FIG. 1. MELVILLE AND MACALPINE'S REDUCTION-GEAR CONNECTED TO WESTINGHOUSE TURBINE

EXPERIMENTAL SPUR-WHEEL REDUCTION-GEAR

THE inherent desirability of interpolating some form of reduction-gear between a marine steam-turbine and the propeller it drives has long been sufficiently obvious. Suggestions for effecting this by electrical or hydraulic means have been frequent, and, in fact, we hope to illustrate shortly an hydraulic coupling which has given on trial really remarkable results. The straightforward solution of the problem by means of a pinion and a spur-wheel has generally been dismissed as impracticable, although the remarkable efficiency of a well-constructed toothed-wheel drive is known to all. The waste of energy in such a transmission gear is notably less than is commercially practicable with an electric coupling, which in turn is superior in this respect to the hydraulic system. Nevertheless, the fact that turbines of the Laval type were not constructed to transmit more than some 300 horse-power constituted *prima facie* evidence that with higher powers special difficulties were to be feared. The remarkable results obtained with motor-car drives, where tooth pressures at times exceed 2,000 lb. per linear inch, were no sure indication that equally good results could be obtained in transmitting by gears some 4000 horse-power instead of 40, since the rule of three is by no means always applicable in practical constructive engineering. Nevertheless, in his lectures on the steam-turbine before the Society of Arts last spring, Mr. Gerald Stoney did suggest that the newer alloy steels and the greater accuracy of workmanship, which have made gearing so successful in motor-car practice, did hold out a prospect that the problem of the marine steam-turbine might be solved by means of ordinary toothed reduction-gearing.

Those interested in steam-turbines have known for some months past that a solution of the problem on these lines was, with his characteristic enterprise, being put to a practical test by Mr. George Westinghouse, who was having constructed for him a double-helical spur-gear intended to transmit 6000 horse-power at 1500 revolutions of the pinion per minute, the reduction ratio being about 5 to 1. Through the courtesy of Mr. Westinghouse we are now able to publish illustrations of this remarkable experimental gear, the details of which have been worked out by Rear-Admiral George W. Melville and Mr. John H. Macalpine, who have patented the novel features on which they rely for success. A perspective view of the gear, with the casing partly broken away, is represented in Fig. 1, on the opposite page, which shows the device coupled to a Westinghouse double-flow turbine. The forgings for the gears were obtained from Messrs. Krupp, of Essen, the teeth being

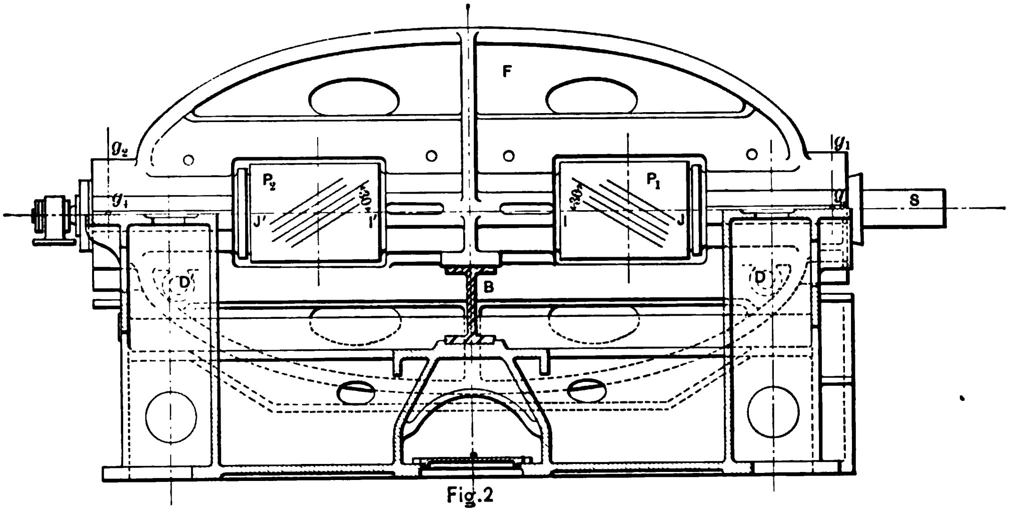


Fig.2

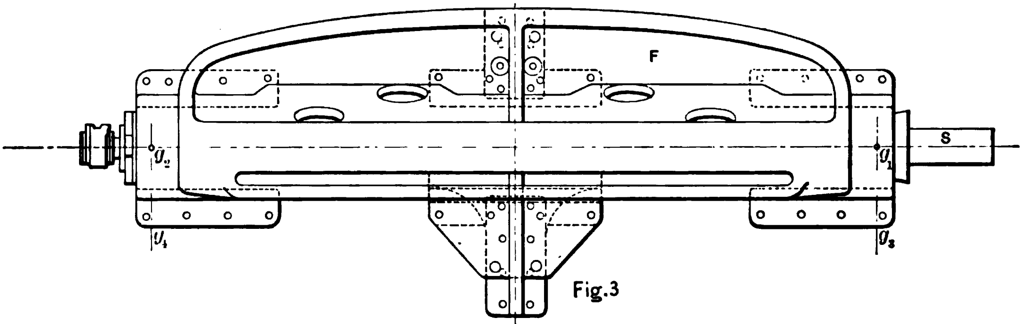


Fig.3

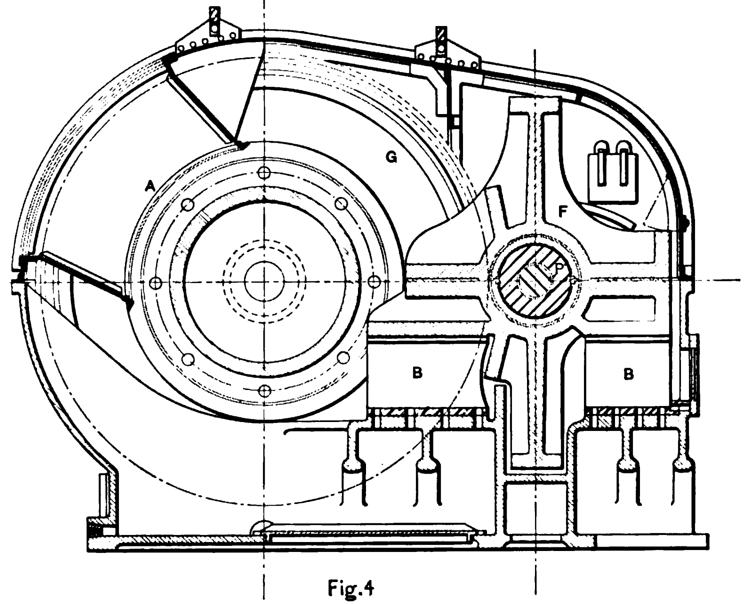


Fig.4

cut by Messrs. Shuchardt and Schütte, Chemnitz, whilst the standing parts were made by The Westinghouse Machine Company.

The pinions have 35 teeth each and the spur-wheels 176, a hunting-cog being introduced to equalize wear. The pitch is 1¼ in., and the pitch helices are at an angle of 30 deg. with the axis of the shaft. One wheel and pinion have, of course, right-handed helices, and the other pair left-handed, so as to eliminate end-thrust. The diameter of the pitch circle of the large wheels is about 70 in., and of the pinions 14 in. Details of the gear-box are shown in Figs. 2, 3 and 4 on the opposite page. These have been designed after a long and comprehensive study of the circumstances which tend to interfere with the proper working of very broad teeth. A small pitch was deemed essential if a reasonable absence of noise was to be secured, and this necessarily meant broad teeth, in view of the fact that 6000 horse-power was to be transmitted with a pitch-line speed of very nearly 100 ft. per second, and with a limiting pressure of 453 lb. per lineal inch on the teeth.

In order to ensure the proper working of such broad teeth, Messrs. Melville and Macalpine have adopted the plan of mounting the pinion shaft in a device which they have dubbed a "floating frame," the object of which is to cause the alignment and position of this shaft to be controlled wholly by the interaction of the teeth in contact, and not by the greater or lesser skill of the workman in laying out and fitting the bearings, which, moreover, even if exactly right to start with, could not be depended on to maintain permanently their alignment. The "floating frame," which is lettered F in Figs. 2 to 4 on the opposite page, is a heavy steel casting flexibly mounted in the gear-box and supporting in rigid bearings the pinion shaft, but in such a way as to allow of this shaft having a slight longitudinal freedom, so that it can slide axially to and fro within the frame.

Fig. 2 is an elevation of the floating frame showing the flexible **I**-beams B, on which it is supported from the bed-plate. The pinion, it will be seen, is supported in three bearings, the centre bearing dividing the tooth faces P_1 and P_2, of which P_1 is right-handed and P_2 left-handed. The large gear, being much stiffer, requires only two bearings. The floating frame is made very deep vertically, so as to be exceedingly stiff to withstand deflection by the nearly vertical forces at the bearings. It is also amply stiffened in the horizontal plane to obviate deflection from the weaker horizontal forces. The floating frame is shown in plan in Fig. 3, whilst Fig. 5, on the following page, shows the pinion shaft and the flexible shaft S by which it is driven. At the end of this shaft is a coupling C driven by the turbine-shaft T. The axes of the gear and pinion are horizontal and parallel to one another when everything is in correct adjustment. But it will be evident that the **I**-beams B are virtually a hinge, and if the large gear G was not in place, or the coupling C connected, the axis of the floating frame and pinion could easily be deflected in the vertical plane through a small angle by a slight flexure of the webs.

The coupling consists of two flanges C_1 and C_2, mounted on the shafts T and S, and connected by two transverse links L_1 and L_2, and by a centre pintle, care being taken not to make this restraint redundant. The shaft T can, therefore, only rotate the shaft S through the links L_1, L_2, but as these are transverse, no longitudinal forces can be transmitted. Even when seriously out of adjustment, the longitudinal forces are so small as to be negligible. The pinion, therefore, has perfect freedom of longitudinal movement in its bearings. Further, the pinion is driven by the shaft S, which passes completely through it to the end distant from the coupling, where it is keyed and bolted. This shaft is so flexible that it imposes practically no constraint on the pinion and floating frame such as would prevent slight angular yield of the **I**-beams B. It is evident, therefore, that, both as to its longitudinal position in the floating frame and in the angular position of its axis, the pinion is solely under the control of the forces transmitted by the teeth of the large gear.

Fig. 5

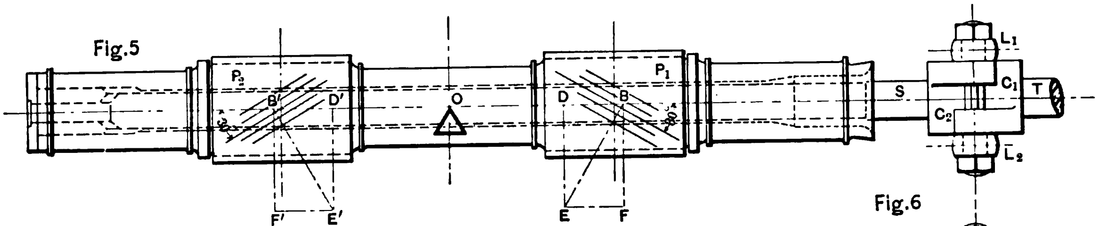
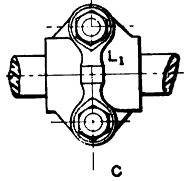


Fig. 6



Neglecting the very slight friction of the well lubricated teeth, the total forces at the tooth contacts, indicated by B, E and B^1, E^1, Fig. 5, will be nearly at right angles to the teeth at B and B^1; that is, they will be at 30 deg. to the vertical. Hence the two parallelograms of forces shown will be similar. Now the horizontal forces at B D, B^1 D^1 are the only axial forces acting on the pinion. If these are not equal, the pinion will at once shift longitudinally, which it is free to do, till they are equal. The two parallelograms of forces then become equal to one another in every respect, and the vertical force B F is equal to the vertical force B^1 F^1. But besides this longitudinal movement of the pinion, the frame in which it is mounted is free to rotate about the centre point O by flexing the **I**-beams. Hence it follows that the moment of the vertical force B F about O must be equal to that of B^1 F^1 about the same point; and as B $F = B^1$ F^1, the arms of these levers O B and O B^1 must be equal. Thus it is claimed that not only will the total forces on the teeth P_1 and P_2 be equal, but the distribution will be so similar that the centres of pressure B and B^1 will be similarly placed.

This, it is contended, practically assures a good distribution of pressure under all conditions of load. For, by sufficient running of the gear, it may readily be assured for one particular load; and if the pinion and floating frame are made sufficiently stiff, this condition will not be seriously changed by increase or decrease of load. The strength of both pinion and floating frame are far in excess of that requisite to sustain the maximum forces to which they will be subject, and their dimensions were made such as to give ample rigidity. For instance, under full load the flexure of the cast-steel floating frame in the vertical plane is so slight that the end bearings will be lowered relatively to the centre by not more than $\frac{1}{2000}$ in.

There is, moreover, a certain compensation for this deflection, owing to the action of the lubricant. Professor Osborne Reynolds's researches on the action of the lubricating films in Mr. Beauchamp Tower's well-known experiments on lubrication seem clearly to indicate that at 1500 revolutions per minute of pinion in the present gear the oil-film in the bearings will be well over $\frac{1}{1000}$ in. thick. When the load on the gear is increased, the end-bearings will be forced down, bending the pinion and making the teeth bear harder at the inner ends I and I^1. Thus more than the proper force will be exerted on the centre bearing, and less on the outer bearings. This will at once lead to a thinning of the oil-film on the centre bearing, and to its thickening on the outer bearings. As the total thickness of the oil-film is more than twice the yield of the outer bearings, this change in the thickness of the film will go far to compensate for the elasticity of the floating frame.

Given that the axes of the gears are parallel to start with, the action of the teeth will, it is claimed, force the axis of the pinion into exact alignment with that of the large gear, provided that in the flexure of the I-beams supporting the floating frame the pinion shaft tilts in a vertical plane parallel to that containing the axis of the large gear. If there were no floating frame, and all bearings of the pinion and gear were cast in one bed-plate, the longitudinal dimensions remaining unchanged, an error of alignment of the axes in the vertical plane not exceeding $\frac{1}{1000}$ in. in the length over the bearings would entirely upset the uniform distribution of the pressure along the faces of the teeth. Hence, with the most careful work, the above assumptions may not be fulfilled with sufficient accuracy. Even if they were exactly fulfilled when the gear was first completed, inequality of the side wear of the bearings might cause the axes to stand at a sensible angle in the horizontal plane.

Provision is made for measuring such errors of alignment as follows:—Outside the end bearings of both pinion and gear, as shown at g_1, g_2, g_3, g_4 in Figs. 2 and 3, there are vertical and horizontal gauges placed so that errors of alignment can be readily measured when the gear is running. If the horizontal gauges at g_3, should show the axes too close by, say, $\frac{1}{10}$ in., and those at g_4 should show them too open by $\frac{1}{10}$ in., giving a total error of alignment of $\frac{1}{5}$ in. in the length between the gauges, it may be shown that, with involute teeth, which are here used, they will bear hard at I and I^1, Fig. 2, and lightly at J and J^1. If the teeth were perfectly rigid, there would, of course, be point contacts at I and I^1, and the teeth would actually stand open by a small amount at J and J¹. This opening Messrs. Melville and Macalpine call the "opening of contact."

If in changing from perfect alignment the pinion turned about either a vertical or a horizontal transverse axis, it may readily be shown that the opening of contact would be a considerable fraction of the sum of the errors shown by the end gauges. On the other hand, the inventors state that the action of the floating frame in combination with involute teeth is such as to constrain the pinion to turn about an axis parallel to what is known as the line of action of the teeth—in the present case, an axis inclined 14 deg. 30 min. to the vertical—and this is precisely the movement which reduces the opening of contact to a minimum.

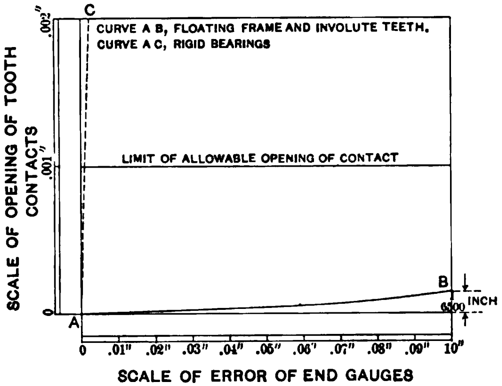


FIG. 7

The result is shown in Fig. 7. The error of each end gauge is indicated by the horizontal scale, the total length representing $\frac{1}{10}$ in., corresponding to a total error of alignment in the horizontal plane of $\frac{1}{5}$ in. between the gauges, as explained above. The vertical scale shows the "opening of contact" if the teeth are rigid, this scale being enormously magnified, as the quantities to be represented are so small. The curve A B represents the resultant opening of contact for given gauge errors for the present gear. The extreme opening of contact shown is $\frac{1}{6,500}$ in., and this for a gauge error which should never be approached. At half this gauge error, still large and easily measurable, the opening of contact is reduced to one-quarter of the foregoing, or $\frac{1}{25,000}$ in., a quantity far below the limit of accuracy of the very finest machining.

Consider next the effects of the elastic strains of the teeth and shafting. Steel gears of 1¼ in. pitch are commonly run under a load of 1000 lb. per inch of tooth face at ordinary pitch-line speeds. As all gears are liable to errors of alignment, this average pressure actually means that at times there will be a maximum pressure very much greater. In the gear under discussion the highest mean value used is less than 500 lb. per inch of tooth contact. Suppose then that by errors of alignment the actual value varied from zero to 1000 lb. per inch, which is far within the disturbance occurring in ordinary gears. It may be shown that the yield of these teeth under 1000 lb. per inch is such that the distribution of pressure would close up an opening of contact of nearly $\frac{1}{1000}$ in. Thus, even under the extreme gauge errors shown at B, Fig. 7, Messrs. Melville and Macalpine claim that the disturbance of distribution of contact pressure is comparatively slight, and at half the gauge error stated it would be insensible. They claim therefore that gauge errors of sensible magnitude do not produce a sensible disturbance. Errors of erection will be exceedingly minute, so that the only errors to be avoided are those due to unequal horizontal wear of the bearings. Any unequal vertical wear will be compensated for by slight tilt of the floating frame. As the forces on the floating frame are inclined to the vertical at only a small angle, vertical wear will take place much more quickly than horizontal; and as the inequality of the wear will usually be but a small fraction of the total wear, it would be almost impossible for any very sensible angular error in the horizontal alignment, such as assumed above, to creep in before the bearings were worn down a long way vertically, and ready for renewal. The case they state is very different if the floating frame is dispensed with, and all the bearings are cast in one bed-plate. The worst supposition is that there is an error of alignment of the axes in the vertical plane. Keeping all dimensions of the gear and bearings the same as in the gear already built, the resulting "opening of contact" is, they find, shown by curve A C, Fig. 7. This exhibits extreme sensitiveness to this error of alignment, as allowable limits will be passed before each vertical gauge shows an error of $\frac{1}{1000}$ in. Such an error would be so small as to be quite insensible to the eye, and would arise even from strongly screwing down one of the holding-down bolts, or from slight heating of the bearings, and could not be avoided by the most perfect workmanship. Hence the gear with rigid bearings they hold to be quite impracticable.

Involute teeth have been used, as they give the excellent results shown by the curve A B, Fig. 7. But they have also the well-known property that if, through wear of the bearings or errors in the original setting up, the axes are somewhat farther apart than designed, the teeth still run true, and there is no "opening of contact." This does not hold with any other form of teeth. For instance, with epicycloidal teeth the action will not be true if the centres move apart, the axes remaining parallel; and in these fine high-speed gears a recession of the centres not exceeding $\frac{1}{1000}$ in. would, the designers assert, be quite serious. With this form of teeth also, when the alignment is defective, an opening of contact of $\frac{1}{1000}$ in., supposing the teeth rigid, corresponds to an error of each end gauge of $\frac{1}{100}$ in.—that is, a total error of alignment between the gauges of $\frac{1}{50}$ in., which could hardly be detected. Besides, on account of the way in which openings of contact occur with epicycloidal teeth, the limit of allowable opening of contact is, it is claimed, much under $\frac{1}{1000}$ in. Hence this form of tooth is far inferior to the involute on account of its extreme sensitiveness to errors of alignment, and also from the fact that it will not run properly when the axes part even slightly, so that the stresses would be much intensified, and the action of the teeth would be noisy. The curve A B (Fig. 7) is tangential to the base-line at A, while for all other arrangements and forms of teeth the curve of opening of contact cuts this base-line at an angle. Hence the solution presented by the floating frame with involute teeth is unique.

From the foregoing it will be seen that the elasticity of the I-beams allows freedom of movement of the axis of the pinion and floating frame in the vertical plane. But by warping the webs of these beams the axis of the floating frame might also turn slightly in the horizontal plane. It is, of course, many times more rigidly mounted against this displacement than against that in the vertical plane, but still the rigidity is by no means complete. Now it may appear that this is not specially important, as from curve A B, Fig. 7, it would seem that such an angular displacement in the horizontal plane, even if of measurable amount, causes no important disturbance. But that is not the only consideration, as the question then arose as to whether with freedom of rotation both in the vertical and horizontal planes the position of the axis of the floating frame would be stable or unstable; that is to say, if forcibly displaced from exact alignment to a small angle in the horizontal plane, will the forces at the tooth contacts tend to bring it back to perfect alignment or to further displace it? By consideration of the slight changes of direction of the forces at the tooth contacts produced by such displacement, it can readily be shown that, with involute teeth, the position of the axis of the floating frame is unstable, and a slight displacement arising from any cause will tend to increase. With the **I**-beams alone, therefore, there might arise excessive horizontal errors of alignment, and trifling causes might produce sudden and violent changes of the horizontal error, either of which effects would endanger the whole mechanism. The floating frame has therefore been deprived of all freedom of motion in the horizontal plane by means of two horizontal struts standing transversely between the floating frame and bed-plate. These bear on the floating frame at D and D^1, Fig. 2, and one is clearly shown in Fig. 1. Obviously these struts in no way interfere with the freedom of movement of the floating frame in the vertical plane. Where they bear in the bed-plate there is an adjusting and locking mechanism which greatly facilitates the true setting up of the floating frame, and is to be seen in the perspective view in the front of the bed-plate, near the end next the turbine.

With epicycloidal teeth in perfect adjustment, the position of the floating frame axis can be shown to be stable without struts. But this is only an apparent advantage, as this stability would at once be upset by widening the gear and pinion centres, or by the slightest wear. But even if this stability could be trusted, it is believed that epicycloidal teeth are far too sensitive to errors of adjustment to be used, and the question of stability or instability in no way affects this consideration.

The only condition which could cause serious disturbance of the tooth pressures is an excessive heating of the pinion above the temperature of the gear. But the design provides for a copious application of lubricating and cooling oil, especially to the pinion, which has most tendency to heat. The cover, also, is so arranged as to draw in air at the ends and at A, Fig. 4, by the fan action of the gears, and discharge it by openings (not shown) to the right of this cross-section. Besides, water can be circulated, by means only partly shown in the figures, between the pinion and the flexible shaft S. But it is not anticipated that the heat to be removed per minute will be so great as has been provided for. The gears of heavy electric trains work under much higher stress both in the metal and between the teeth in contact—very much higher, as these stresses are accentuated by errors of alignment shown by rounding of the teeth through wear—and with practically no lubrication. Yet they run for long distances. In the reduction-gear these unfavorable conditions are reversed—there is good lubrication and very uniform contact. If the coefficient of friction at the tooth contacts is $\frac{1}{10}$, it can readily be shown that the frictional loss at the teeth will be under 1 per cent. So that even if the gear were transmitting 6000 horse-power—the very highest power hoped for—this frictional loss would be under 60 horse-power. As at least one-half will go to the large gear, where it will readily be dissipated from the large surface, there is left well under 30 horse-power to remove from the pinion, which, with the various means provided, should present no difficulty. If then this control of the heat is assured, there will, it is claimed, be practically perfect tooth contacts; this, with the fine pitch and the spiral gears (avoiding sudden entering or leaving contact of one whole tooth at a time), will, it is asserted, ensure reasonably quiet running of the gears.

At 1500 revolutions per minute of pinion and 453 lb. per inch of tooth contact, the gear will transmit 6000 horse-power. The pitch-line speed would then be 5500 ft. per minute, and the mean speed of sliding about one-tenth this, or 550 ft. per minute. There is on record a case of a right-angled worm-gear run successfully at 15,000 ft. per minute with a contact pressure of 350 lb. per inch. The average speed of sliding will here be greater than the pitch-line speed by over 41 per cent., or, say, 21,000 ft. per minute. Taking speed of sliding multiplied by pressure as a rough means of comparing these two cases, we get:—

$$\frac{\text{Reduction-gear . . . . . .}}{\text{Right-angled worm-gear . . .}}\ \frac{5500 \times 453}{21{,}000 \times 350} = \frac{1}{29.5}$$

The extreme constants chosen, therefore, seem well within practical limits.

Before the experiment is tried it is impossible to predict the limit of safe load, but the following are two cases which it will be interesting to consider:—

1. At 1000 revolutions per minute of pinion, and 283 lb. per inch of tooth contact—that is, as shown above, only 28 per cent. of the intensity of pressure used with steel gears of 1¼ in. pitch at ordinary pitch-line speeds—the gear would transmit 2500 horse-power. The famous merchant ship Aberdeen, by the engines of which Dr. Alexander C. Kirk introduced triple-expansion, was about of this power. Her main engines, without shafting, propeller, stern-tube, spare gear and fittings, weighed 221 tons. If we deduct from this the condensers, pipes, sea-valves, air-pumps, circulating pumps, donkey pumps, floors and gratings, etc., the weight

left for the main engines proper will be about 150 to 160 tons—say, 150 tons. Hence the following comparison may be made:—

Westinghouse marine turbine of 3000 brake horse-power, 1000 revolutions per minute, weighs	27	Tons
Reduction-gear	25	"
Total	52	"
Aberdeen's engines replaced	150	"
Saving	98	"

That is, 65 per cent. of the weight is saved. If this design will transmit 2500 horse-power, it is claimed that similar gears could be applied with great saving of weight and space to all steamships up to 18 knots, which comprise over 95 per cent. of all the ships afloat.

2 As already stated, at 1500 revolutions per minute of pinion, and 453 lb. per inch of tooth contact (or only 45 per cent. of the intensity of pressure for similar teeth at ordinary pitch-line speeds), the gear will transmit 6000 horse-power. If this be realized, similar gears could be applied to any speed of ship. This design would then be of the correct power for the Dreadnought, and applied to her would save fully 50 per cent. of the weight of her turbines, besides reducing boiler weights, since both turbines and propellers would now be of considerably enhanced efficiency.

Increasing, by the law of comparison for similar machines, the dimensions and power of the present design, from 6000 horse-power, a size is obtained suitable for the Mauretania with three large screws of the same total power as the present four-screw ship. Here again the weight of the turbines would, it is claimed, be halved, as also the engine-room length. The boilers also, as in the Dreadnought, would, it is considered, be materially reduced.

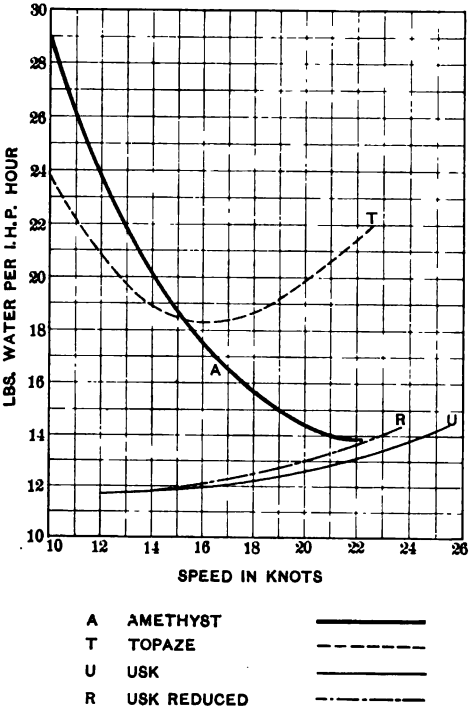


FIG. 8

In the plate on pages 24 and 25 the upper views represent the Mauretania thus modified, as compared with the existing arrangement shown in the lowest of the three views. For warships the saving effected by such a gear should, it is claimed, be considerable, especially at cruising speeds. In Fig. 8 is reproduced a curve A, showing the coal consumption of the Amethyst as compared with the Topaze (curve T). The latter, however, was not fitted with reciprocating engines of maximum efficiency, much better results being obtainable where the conditions are better suited to the reciprocating engine. The curve marked U shows the consumption of the Usk on the assumption that at full power the evaporation was 8 lb. of water per pound of coal, and at 13 knots 9 lb. The curve lettered R is the Usk's curve reduced to a smaller scale of knots, so as to make the full speed the same as that of the Amethyst. On these assumptions it will be seen that the reciprocating engine had the advantage, but, in justice to the turbine, it must be added that much better results have been obtained with the newer turbine cruisers. Nevertheless, the fact remains that marine turbines are commonly under-speeded.

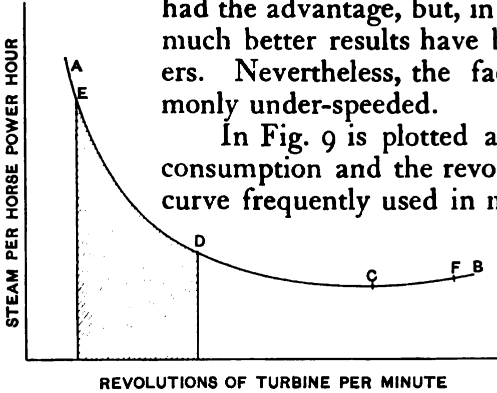


FIG. 9

In Fig. 9 is plotted a curve showing the relation between steam consumption and the revolutions of the turbine. The portion of the curve frequently used in marine practice is hatched, and shows clearly how very rapidly the efficiency falls off as the speed diminishes. By using gearing it is contended that the turbine at maximum power can be even slightly over-speeded, as at F, so that at cruising speeds the efficiency will actually be increased.

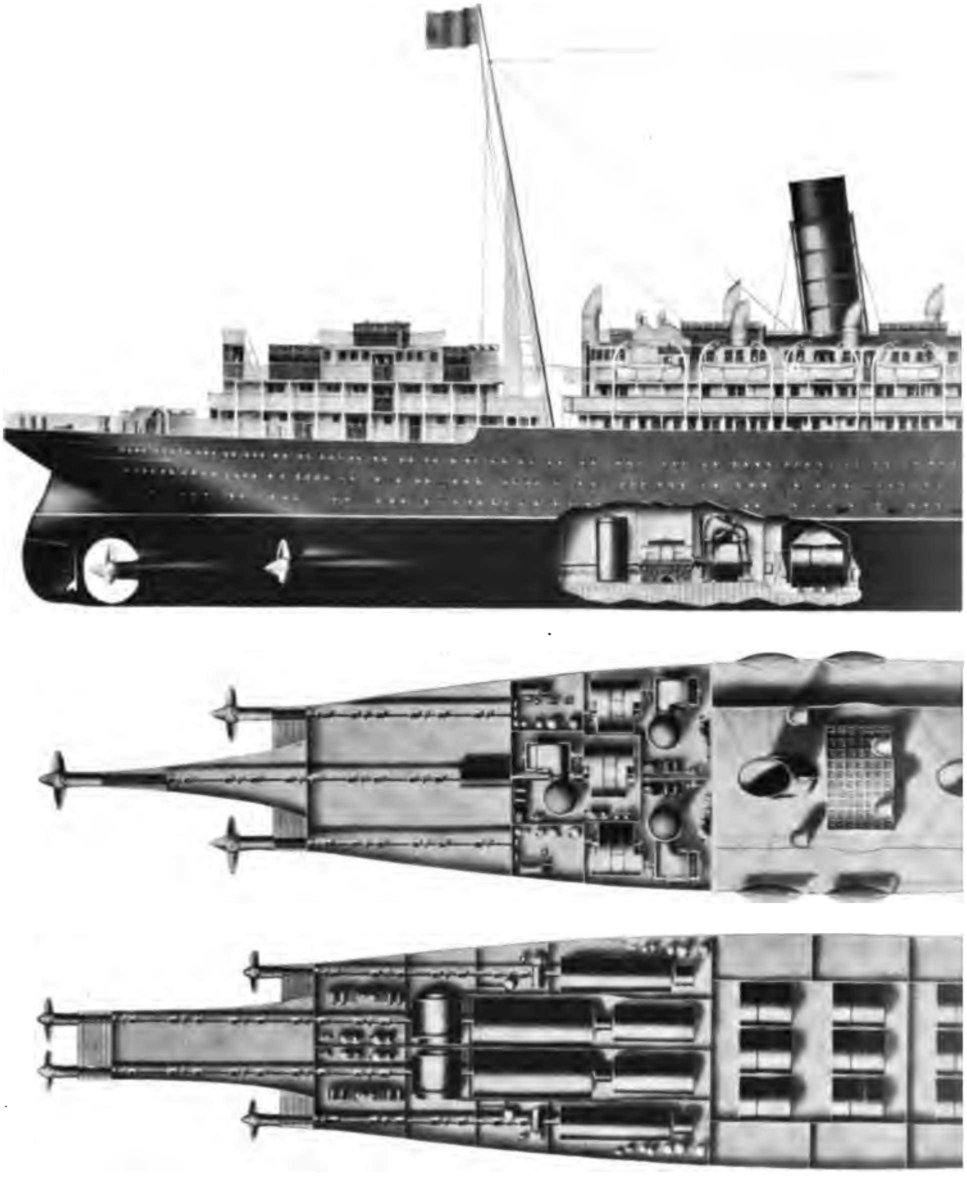
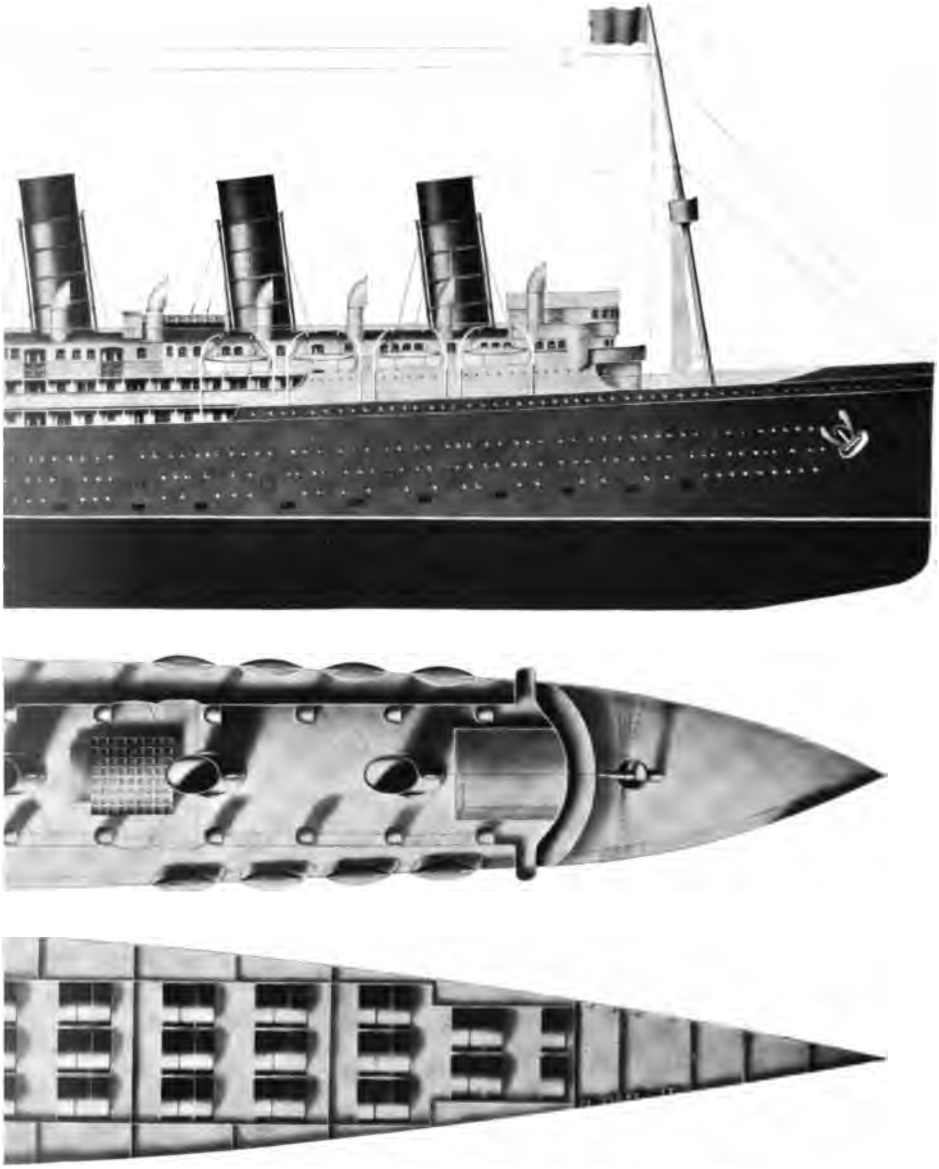


DIAGRAM ILLUSTRATIVE OF RELATIVE SPACES
TURBINES IN SS



CUPIED BY GEARED AND DIRECT-DRIVING
IAURETANIA

6,000 HORSE-POWER HYDRAULIC ABSORPTION DYNAMOMETER

FOR TESTING THE EXPERIMENTAL SPUR-WHEEL REDUCTION-GEAR AT THE WORKS OF THE WESTINGHOUSE MACHINE COMPANY, EAST PITTSBURG, PA.

6000 H. P. HYDRAULIC ABSORPTION DYNAMOMETER

IN order to carry out the tests to determine the capacity and efficiency of the Experimental Spur-Wheel Reduction-gear for High Speed Turbines, constructed by The Westinghouse Machine Company, East Pittsburg, Pa., U. S. A., after the designs of Rear-Admiral George W. Melville and Mr. John H. Macalpine, it was necessary to provide some form of brake or dynamometer that would be capable of absorbing a continuous output of 6000 effective horse-power.

A brake of this capacity was in its way as entirely unprecedented as the gear itself. In any form of mechanical brake, the energy is transformed into heat, and in dealing with such large powers as were contemplated, the problem of carrying away the heat generated becomes a very serious one. When one recalls that the quantity of heat generated by the absorption of one horse-power is 2,545 British thermal units per hour, or about one-fifth that resulting from the combustion of a pound of coal, one is better able to comprehend the magnitude of the problem by making the simple calculation that shows the total quantity of heat to be dealt with, is practically that which would be generated in a furnace burning coal continuously at the rate of 1,200 pounds per hour.

An ordinary band or "Prony" type of friction brake was entirely out of consideration, as the speed would limit the diameter of the brake wheel to six or seven feet, and its face would have to be not less than thirty feet. The difficulties involved in the construction, and the cooling of such a brake, and the maintenance of the proper tension on a brake band of this width are practically insurmountable.

Naturally the first thing to suggest itself was the use of a dynamo electric machine with a water rheostat for adjusting and maintaining the load. Such a generator would be cumbersome and expensive, the windage and electrical losses would vary with different loads and speeds, and there would always be an uncertainty as to the exactness of the allowances made for these losses. Neither are the most careful electrical measurements quite as satisfying and convincing as direct comparisons with the familiar force of gravity.

The idea of the electrical dynamometer was therefore abandoned because of the uncertainty that its indications would be sufficiently exact to enable the transmission loss in the gear to be determined with any reasonable degree of finality, as it was hoped that this loss would be very small, and possibly within the probable limits of accuracy of this method of measurement.

For a number of years The Westinghouse Machine Company has successfully employed a form of hydraulic brake for measuring the power developed in the works tests made on all steam turbines manufactured by it. This form of brake consists of a rotor mounted on a shaft coupled to the shaft of the turbine, and rotating within a closed casing supported on journals through which the rotor shaft passes. The casing is held against turning by means of a radial arm bearing against a vertical strut, the lower end of which rests on an ordinary platform scale. Within the casing is a quantity of water, which quantity may be maintained constant at any desired amount by means of inlet and outlet pipes with adjustable controlling valves. The rotor has a series of vanes or teeth on its periphery which tend to impart a rotary motion to the water contained in the casing. The inner surface of the casing is provided with vanes or teeth which resist this tendency, and the result is a powerful braking action, the intensity of which can be regulated by the quantity of water in the casing. The resistance which the casing must offer to prevent its being rotated by the water striking its inwardly projecting vanes at high velocity is measured by the pressure exerted on the platform scale by the strut under the radius arm, and from this pressure, the effective length of the radius arm, and the number of revolutions per minute, the power is calculated in identically the same manner as in the case of an ordinary Prony or band brake. As a matter of course, the temperature of the water is quickly raised to the boiling point, and a considerable portion of it evaporates, and in the form of steam easily carries off the enormous quantities of heat generated. The amount of fresh water admitted to compensate for the quantity lost by evaporation and through the overflow pipes, is constantly adjusted by the attendant in such a way as to keep the scale at all times in perfect balance.

The successful brakes constructed on this principle had all been designed for high speeds, ranging from 750 revolutions per minute for the largest, up to 4000 or more for the smallest sizes. In order to adapt the same principle to a brake that should have the requisite capacity and stability at the lower speeds of the driven shaft of the reduction-gear, the radically new design, of

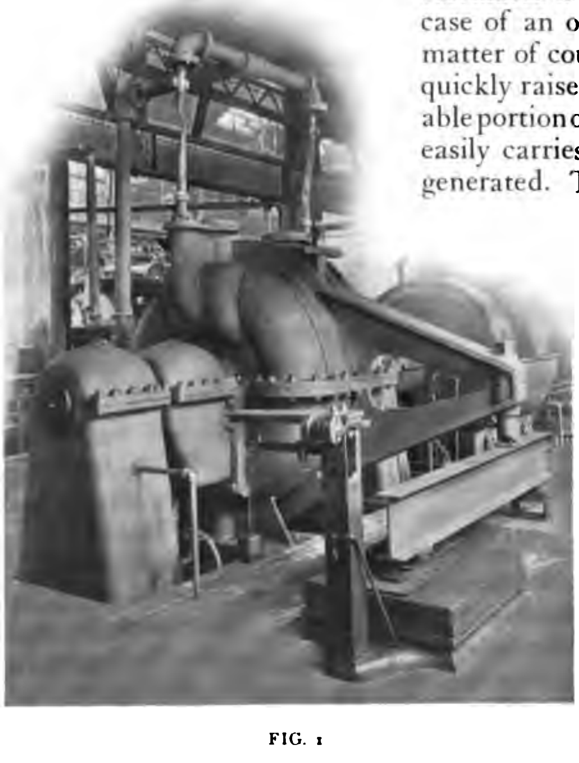


FIG. 1

which a detailed description follows, was developed by Mr. Raymond N. Ehrhart of The Westinghouse Machine Company's Engineering Staff.

Unfortunately, the space between the experimental plant and the wall of the building in which it is located is too restricted to enable a satisfactory photographic view to be made of the complete installation comprising turbine, reduction-gear and dynamometer. The line drawing (Fig. 2), however, shows clearly and correctly the disposition and relative proportions of the several component parts.

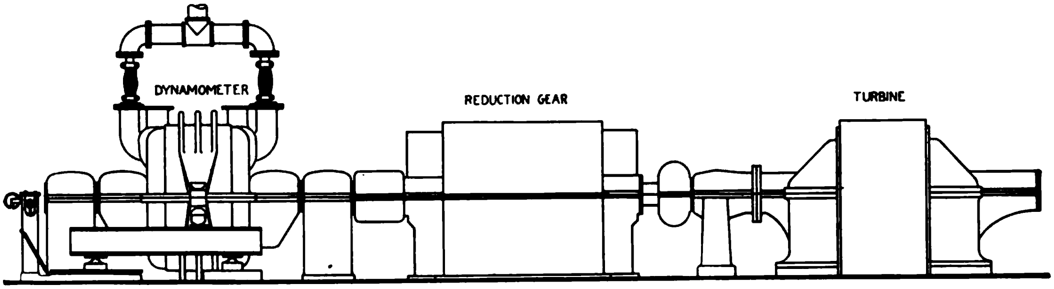


FIG. 2

Figure 1 on the opposite page is a general view of the dynamometer, and the arrangements for weighing the load. On account of the heavy pressures to be dealt with, the resistance of the radius arm is not taken up directly on the platform of the scale, but is transmitted through a knife edge bearing to an **I**-beam, one end of which rests on a solid foundation and the other end on the platform of the scale. The **I**-beam also rests on knife edge bearings at either end, and forms a lever of such proportions that only one-fourth of the total stress comes on the scale. At the top of the dynamometer will be seen the flexible hose pipe connections for admitting the water to the interior of the casing.

Figure 3 is a view of the interior of the upper half of the casing, and shows the stationary vanes which resist the impulse imparted to the water by the rotor. These vanes are inserted and supported in precisely the same manner as the blades in the cylinder of a steam turbine.

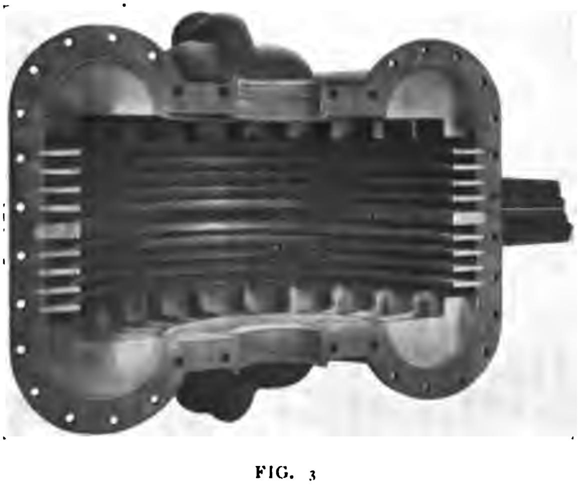


FIG. 3

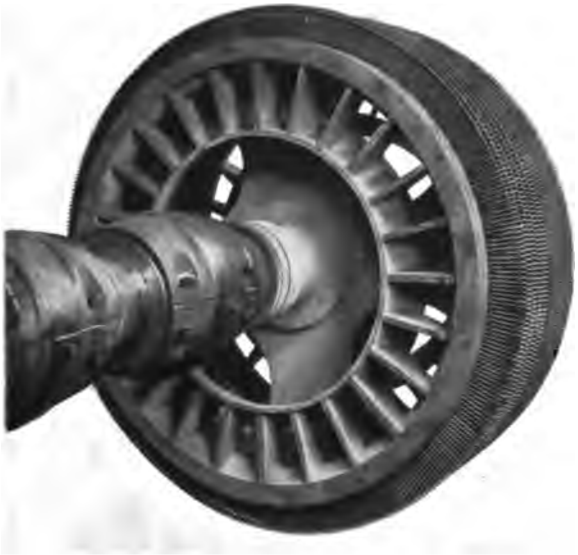


FIG. 4

Figures 4 and 5 are two views of the rotor, bladed with regular steam turbine blading sections, the rows on one side of the center being of what is called "right hand section" and on the opposite side of "left hand section."

The water enters the top half of the casing through ports on either side which register with the passages in the side of the rotor clearly shown in Fig. 4 just inside the rim. Through these passages the water is carried to the middle of the rim of the rotor and is discharged through the ports shown in Fig. 5, between the two innermost rows of blades.

With the aid of the diagram, Fig. 6, on the following page, a developed cross section through the blades, the action that next takes will be readily understood. The moving blades are shown in solid and the stationary blades in cross-lined section. The water emerges with a whirling motion from the port A, and immediately meets the broad central row of stationary vanes which check its angular velocity. It escapes to the right and left from the passages between the central row of stationary vanes and is picked up by the first rows of moving blades on either side. These moving blades again impart a high angular velocity to the water, and by reason of the curvature of their section project it into the adjacent rows of stationary vanes, where the velocity is again checked. This action is repeated as the water passes through the successive rows of moving and stationary blades, until it reaches the outermost rows of moving blades.

From these last rows of moving blades the water is projected into circumferential passages

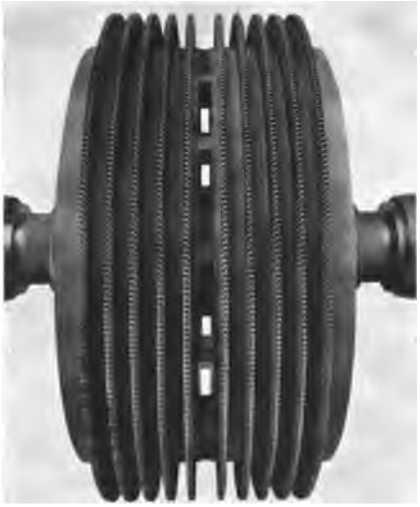


FIG. 5

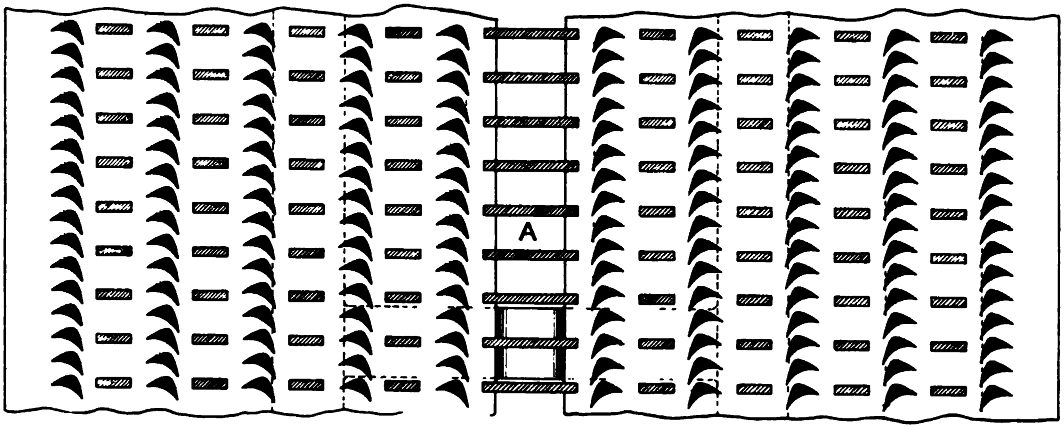


FIG. 6

of semi-circular cross section in either end of the casing.

These passages are shown at B in Fig. 7, a partial cross sectional view through the rotor and casing. At intervals in these circumferential passages are baffles to again check the angular velocity of the water. These baffles are shown plainly in Fig. 3, inside view of upper half of casing. The passages in the casing re-direct the water into the rotor—or at least so much of it as has not been already evaporated—and the entire cycle of operation is repeated indefinitely.

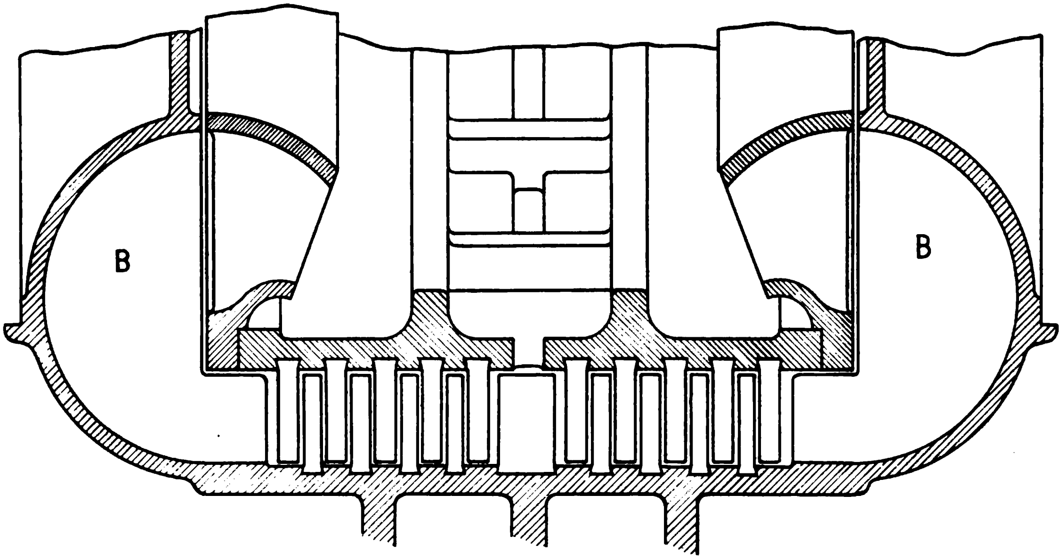


FIG. 7

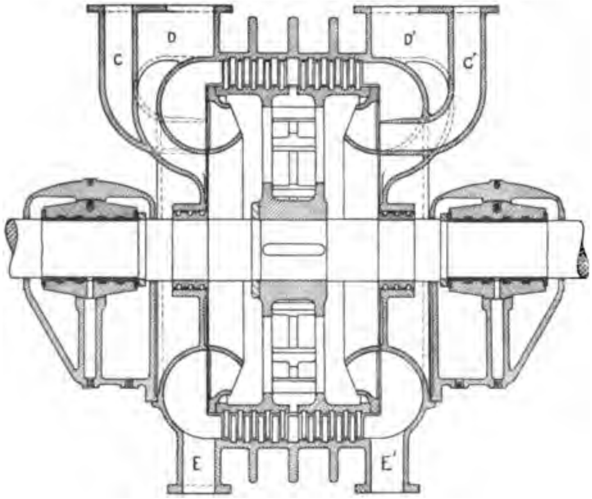


FIG. 8

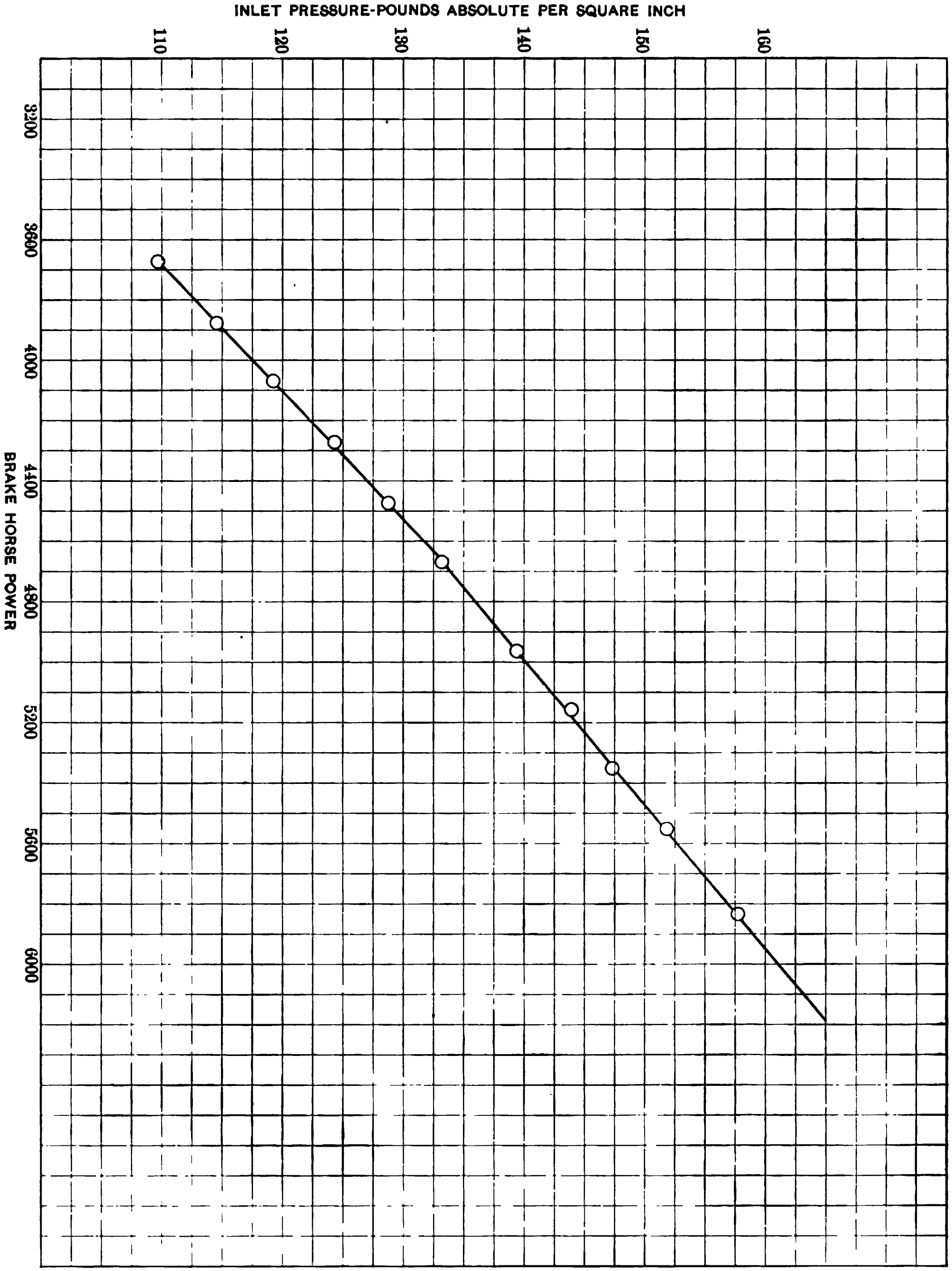
Figure 8 is a section through the assembled dynamometer. The passages through which the water enters are indicated by the letters C and C¹. D and D¹ are vents for the escape of the steam generated by the transformation of the mechanical energy into heat. It is not practicable to carry off all of the heat by evaporation alone, as the generation of 15,000 pounds of steam per hour in the casing would cause such a violent boiling as to interfere with the stability of the braking action. Consequently the quantity of water admitted into the casing is considerably in excess of the quantity evaporated, and the surplus is discharged at boiling temperature through the passages E and E¹.

The dynamometer measures the output of the gear with the utmost precision, but in order to determine the transmission loss, it is imperative that the input should also be known. An extensive series of tests was carried out in which the power was measured by a dynamometer attached directly to the shaft of the turbine. It was found that for each speed and load, with a constant vacuum and with steam of uniform quality, there was a corresponding inlet steam pressure. By often repeated trials it was found that by reproducing any particular set of conditions of speed, inlet pressure and vacuum, the brake load could be predicted with unfailing accuracy. This fact being thoroughly established, it was only a matter of a long and tedious series of routine trials to calibrate the turbine for all of the loads and speeds at which the plant was to be tested. As a variation in the vacuum has the most serious influence on the power of a steam turbine, all other operating conditions remaining the same, the calibration tests were made with an abnormally poor vacuum, so as to allow a more ample margin for control, than would be possible if it had been attempted to maintain a vacuum that would more nearly tax the capacity of the air pump.

The dynamometer, notwithstanding that it was an entirely new creation, has operated in the most gratifying manner as regards steadiness, sensitiveness, and capacity, and in general has creditably upheld its part in what is perhaps the most extensive laboratory experiment that has ever been undertaken and carried out by strictly private enterprise.

REPORT OF TESTS OF THE MELVILLE AND MACALPINE EXPERIMENTAL REDUCTION-GEAR

MADE AT THE WORKS OF THE WESTINGHOUSE MACHINE COMPANY, EAST PITTSBURG, PA.


INLET PRESSURE-POUNDS ABSOLUTE PER SQUARE INCH
110
120
130
140
150
160
3200
3600
4000
4400
4800
5200
5600
6000
BRAKE HORSE POWER

TESTS

WHEN the Melville and Macalpine experimental reduction-gear was first erected and started on September 7, 1909, extensive preparations had been made to determine the efficiency of the apparatus on the assumption that there might be a transmission loss of at least five per cent. After calculating the results of a few preliminary trials, it was found that the apparent efficiency was over 98 per cent., and this raised the perplexing question as to how to determine this unexpectedly small transmission loss with a satisfying degree of exactitude.

If the efficiency were only from 90 to 95 per cent., its measurement, within reasonable limits of accuracy, would have been an easy matter. Rising as it unquestionably does, above 98½ per cent., it will be readily understood that the problem requires a refinement of method, and a degree of painstaking care far beyond that which is usual in ordinary engineering investigations.

For the reason that the efficiency is so remarkable, it has been thought advisable to describe the methods employed and to tabulate the principal observations with a minuteness of detail that might, in many cases, be considered superfluous, if not actually wearying.

Naturally the first trials of the gear were made at very moderate speeds and loads, with frequent stops to examine the condition of the teeth and bearings. During the first few days the speed was gradually raised to 1500 revolutions per minute of the turbine shaft, and 300 revolutions of the gear shaft, and a load of over 6000 brake horse-power was carried. The load was limited only by the power of the turbine, as there were no indications that the ultimate safe capacity of the gear had from any standpoint been nearly approached.

On Saturday, October 16, 1909, an endurance test at maximum load and speed was started at 3.15 p. m., and continued until 7.15 a. m. on the following Monday, or a total of 40 hours. This test was witnessed by Commander R. S. Griffin, U. S. N., and Lieutenant-Commander U. T. Holmes, U. S. N., specially delegated for this purpose by the Bureau of Steam Engineering of the United States Navy Department.

The load was applied by the hydraulic dynamometer described on the preceding pages. The radius arm of this dynamometer is 6¼ feet, and the pressure exerted is transmitted to the platform scale through a lever having a ratio of approximately

1 to 4, as shown in Figs. 1 and 2 on pages 30 and 31 of the detailed description of this piece of apparatus. The precise distances between the knife edges on the lever being, however, 18 inches and $72\frac{5}{32}$ inches respectively, the effective length of the radius arm of the dynamometer is

$$\frac{6.5 \times 72.15625}{18} = 26.056 \text{ feet.}$$

The constant for one pound pressure on the scale, and one revolution per minute is consequently

$$\frac{26.056 \times 2 \times 3.1416}{33000} = .0049612 \text{ h. p.}$$

During this trial the following average conditions were maintained:

Gross weight on scale	4443.4 lbs.
Dead weight on scale	388. lbs.
Net weight on scale	4055.4 lbs.
Speed	300.6 r.p.m.
Brake horse-power	6048.

There was nothing in the operation of the gear to indicate that this load might not be carried indefinitely. It will be readily understood that a constant draft of 6000 to 7000 h. p. on the boiler plant of an industrial establishment like The Westinghouse Machine Company could not be long maintained without interfering with the regular operations of the Company. As a consequence, the duration of an experiment of this nature and magnitude is necessarily limited to the period between noon on Saturday and the early morning of the following Monday, during which time all other activities are in a large measure suspended.

A continuous test of five or six days representing the time of an average transatlantic voyage would doubtless be of more popular interest, but it would be of no more real scientific value. When all the parts of the apparatus have once attained a maximum temperature which remains constant for a reasonable period, and providing that temperature is within the limits generally recognized as conservative, a condition has been established which is capable of being maintained indefinitely. No useful data could be obtained from a further prolongation of the trial, excepting the very probable development that as the wearing parts became polished and took on a better surface, the operation of the apparatus would improve. In the present instance the trial continued over 34 hours after the temperature conditions had become constant.

With the dynamometer described on pages 30 and 31 inclusive, the measurement of the power output of the gear becomes a very simple matter. But to determine at the same time the effective power input, is an engineering problem that is more than a little out of the ordinary. If the gear were driven with a reciprocating engine, the natural procedure would be to measure the indicated horse-power, and either estimate or measure the friction of the engine, and thus obtain a correction which, when applied to the indicated horse-power, would give the effective power delivered to the gear.

There being no way in which to measure the indicated horse-power of a steam turbine, it became necessary to establish the exact brake horse-power in some other way. Happily, there is one characteristic of the steam turbine that makes it possible to calibrate any particular machine, in such a way that its output in effective horse-power at any instant may be determined with greater accuracy than it is possible to determine even the indicated horse-power of a reciprocating engine. As long as the speed and exhaust pressure are maintained constant, the absolute inlet pressure of commercially dry steam, at any instant, is a very accurate measure of the brake horse-power the turbine is developing.

By substituting for the reduction-gear, a dynamometer connected directly to the turbine shaft, and operating the turbine at a fixed speed, and with a constant vacuum in the exhaust pipe, we may determine the inlet pressures corresponding to different loads at this speed.

Plotting these observations on a diagram in which the ordinates represent absolute pressures per square inch, and the abscissae represent brake horse-powers, and drawing a line through the several points, we get a scale by means of which the horse-power corresponding to any inlet pressure may easily be read. This line is practically straight, so that readings made between the points representing actual observations, are quite dependable.

The diagram on page 36 shows the calibration of the turbine used on these tests at 1500 revolutions per minute with a vacuum of 22.78 inches, corrected to a barometric pressure of 30 inches.

The radius arm of the high speed hydraulic dynamometer used in calibrating the turbine is 48.375 inches. The constant per pound pressure on the scale is therefore

$$\frac{48.375 \times 2 \times 3.1416 \times 1500}{12 \times 33000} = 1.1513$$

Table I. on page 40 gives the observed absolute inlet pressures, loads on scale, and calculated brake horse-powers over a considerable range above and below 5000 h. p. From these observations and calculations, a calibration curve similar to that shown in Fig. 1, was drawn to a large scale so that small differences of loads and pressure could be read accurately.

TABLE I.

Inlet Pressure Absolute.	Gross Weight on Scale.	Dead Weight on Scale.	Net Weight on Scale.	B. H. P.
109.6	3585	385	3200	3675
114.8	3755	385	3370	3880
119.3	3925	385	3540	4076
124.3	4095	385	3710	4271
128.8	4265	385	3880	4467
133.3	4435	385	4050	4663
139.3	4690	385	4305	4956
143.8	4860	385	4475	5152
147.3	5030	385	4645	5348
151.8	5200	385	4815	5543
157.8	5455	385	5070	5837

With the reduction-gear connected to the turbine, a number of observations were made of the absolute inlet pressures corresponding to practically the same range of loads transmitted through the gear to the slow speed hydraulic dynamometer.

In these tests, as well as in the turbine calibrations, at every load before reading the inlet pressure, the speed of the turbine was brought exactly to 1500 revolutions per minute as indicated by a vibration tachometer, and the vacuum adjusted to exactly 22.78 inches corrected to a barometric pressure of 30 inches. The vacuum was purposely carried abnormally low by bleeding a small quantity of air into the exhaust pipe. By means of a small valve in the air inlet pipe, the vacuum could be quickly and accurately adjusted to the arbitrary standard without its being necessary to make any change in the speed of the air pump or the quantity of the circulating water.

The gear and pinion having respectively 176 and 35 teeth, the speed of the driven shaft would be 298.3 revolutions per minute. The horse-power constant of the dynamometer is, as stated on page 38, .0049612.

TABLE II.

Inlet Pressure Absolute.	Gross Weight on Scale.	Dead Weight on Scale.	Net Weight on Scale.	BHP Delivered by Gear.	B.H.P. of Turbine from Calibration Curve.	Efficiency.
111.8	2900	388	2512	3712	3771	98.7
122.3	3200	388	2812	4156	4197	99.0
132.3	3480	388	3092	4576	4623	98.9
142.3	3791	388	3403	5036	5108	98.7
152.3	4100	388	3712	5486	5567	98.5
162.3	4400	388	4012	5927	6057	98.7

In Table II. column 1, are the observed absolute inlet pressures corresponding to the different loads transmitted through the gear as set down in column 5. In column 6 are given the horse-powers of the turbine alone corresponding to the same inlet pressures, as determined from the calibration curve. Obviously, the quantities in column 5 represent the power output of the gear, and those in column 6 represent the power input. The quantity in column 5, divided by the quantity standing against it in column 6, gives the efficiency as it appears in column 7.

While the efficiencies as calculated from these observations are consistent and uniform over the whole range of observations, the total transmission loss as shown, almost falls within the limit of the normal error that might reasonably be expected in measuring such large total quantities of power. And in spite of the fact that the greatest precautions were taken to insure accuracy, the readings being checked by several observers, and the gauges and scales being standardized before and after each trial, one would feel some hesitation in vouching for these efficiencies if there were no way of checking them. Fortunately, a satisfactory check is available.

The gear is lubricated by circulating a copious supply of oil under a head of about 10 pounds, through the bearings, and through a spray which plays continuously on the teeth of the wheels. The transmission loss in the gear therefore appears as heat in the oil. The heated oil coming from the gear is passed through a cooler which is very like a surface condenser with cold water circulating through the tubes.

By measuring the quantity of oil circulated and noting the rise in temperature in passing through the gear, a close approximation of the number of British Thermal Units lost per hour in friction is obtainable.

At a load of 5088 h. p. delivered by the gear, 591 pounds of oil were circulated per minute, with an average rise in temperature of 9.86 degrees Fahr. The specific heat of this oil is 0.47, and consequently the total heat absorbed per hour is 591 x 0.47 x 60=164208 B.T.U.

As 2545 B.T.U. per hour is the equivalent of a horse-power, the total heat accounted for in the oil is

$$\frac{164208}{2545} = 64.17 \text{ b.h.p.}$$

The brake horse-power being 5088, and 64.17 horse-power being accounted for by the heat in the oil, the efficiency by this method of calculation would be

$$\frac{5088}{5088 + 64.17} = 98.75\%$$

This figure is probably a little too high as there is an indeterminate small quantity of heat radiated from the gear casing and from the oil piping that has not been accounted for. Even if this were 20 per cent. of that accounted for, which is most improbable, the efficiency indicated would still be 98.5 per cent.

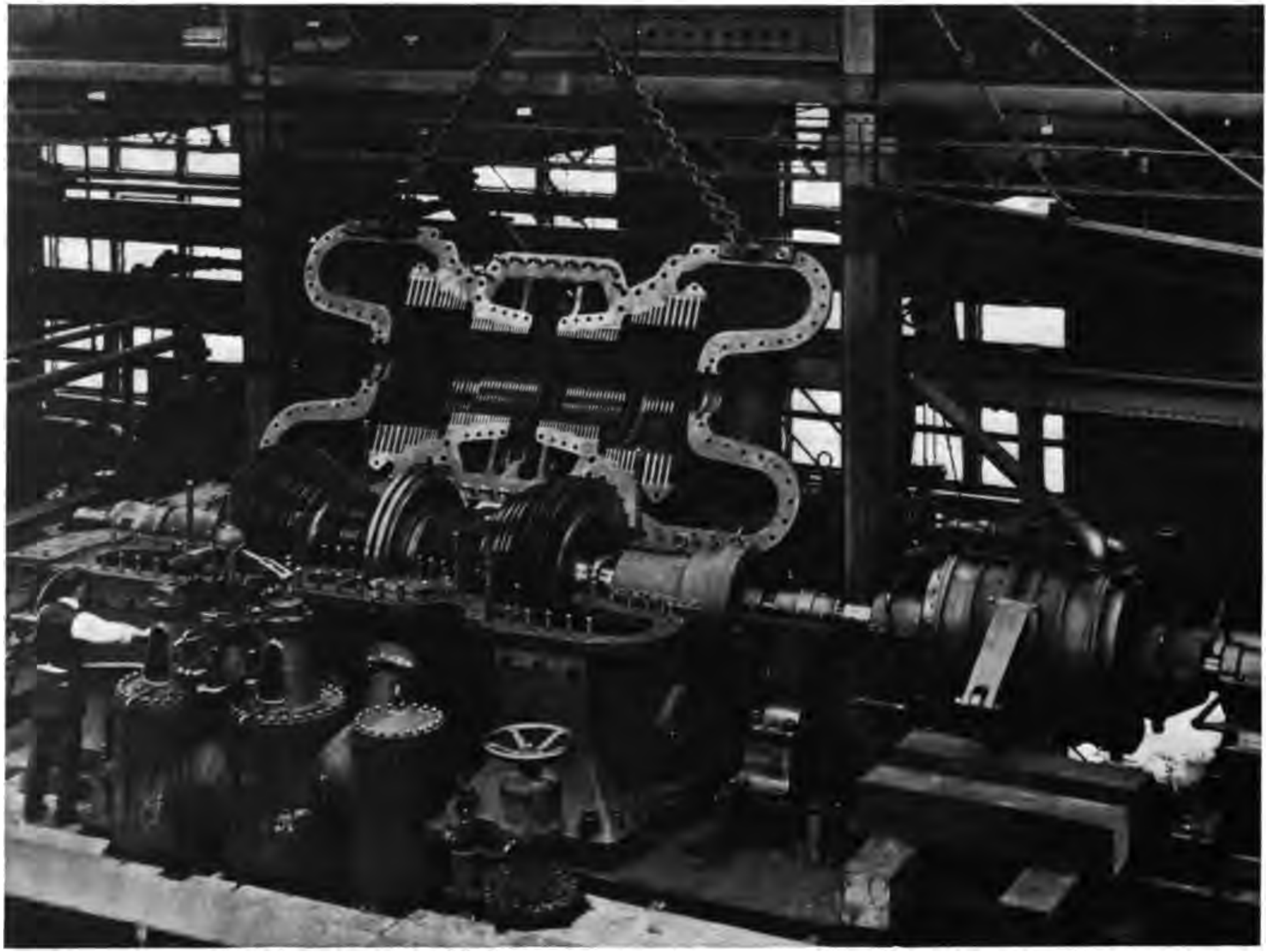
Giving due weight to both methods of measurement, and considering the closeness of agreement between them, it is quite evident that at approximately 5000 h. p., with the turbine running at 1500 revolutions per minute, the efficiency of the gear is more than 98.5 per cent.

Less elaborate and extensive determinations show an efficiency of 98 per cent. at a load of 3096 h. p. and 1000 revolutions per minute of the turbine, and 95 per cent. at a load of only 824 h. p. and 750 revolutions per minute of the turbine.

H. E. Longwell,
Consulting Engineer.

EFFICIENCIES OF HIGH SPEED STEAM TURBINES

AS BUILT BY THE WESTINGHOUSE MACHINE COMPANY, EAST PITTSBURG, PA.



WESTINGHOUSE 22,000 H. P. DOUBLE FLOW TURBINE ON TESTING FLOOR, OPEN FOR INSPECTION

TURBINE EFFICIENCIES

THE Melville and Macalpine Reduction-gear, as has already been fully explained, permits the use of high speed turbines, the proportions of which have been developed to produce extraordinary results.

The engraving on the opposite page illustrates a Westinghouse Combination Double Flow Turbine about to be tested. This turbine is capable of developing 22,000 h.p. with 175 pounds steam pressure and 28 inches vacuum, and it is estimated that the steam consumption will be about 10 pounds per brake horse-power hour. The principal dimensions are as follows:

Length over all	19 feet 8 inches
Height	9 " 0 "
Width	9 " 0 "
Weight	110,000 pounds
Speed	1,800 r.p.m.

This particular machine is designed for land work, and in a modification of the general design to make it especially adaptable for marine propulsion, the weight could be materially reduced.

The following authentic tests lately made on turbines built by The Westinghouse Machine Company show that under the conditions of steam pressure, superheat and vacuum as specified by the United States Navy Department for the battleships "Delaware," "North Dakota," "Utah," etc., a shaft horse-power can be developed on less than 11 pounds of steam per hour.

(1) Westinghouse Combination Double Flow Turbine, No. 563, at the Williamsburgh Power Station of the Transit Development Company, Brooklyn, Speed 750 Revolutions per Minute:

Steam pressure at throttle	203.4 pounds
Superheat	80.1 degrees F.
Vacuum, referred to 30" barometer	28.6 inches
Load	13,384.0 kilowatts
Steam per kilowatt-hour	14.4 pounds
Efficiency of generator	98.0 per cent.
Windage	2.0 per cent.
Equivalent brake horse-power	18,620.0
Steam per brake horse-power hour	10.3 pounds

(2) Westinghouse Parsons Turbine No. 253 at the Waterside Power Station of the New York Edison Company, Speed 750 Revolutions per Minute:

Steam pressure at throttle........	177.6	pounds
Superheat......................	96.0	degrees F.
Vacuum referred to 30″ barometer	27.3	inches
Load...........................	9,830.5	kilowatts
Steam per kilowatt-hour.........	15.18	pounds
Generator efficiency............	98.0	per cent.
Windage.......................	2.0	per cent.
Equivalent brake horse-power....	13,700.0	
Steam per brake horse-power hour.	10.89	pounds

A CRITICISM AND THE REPLY

REPRINTS FROM
"ENGINEERING NEWS"
NEW YORK, NOVEMBER 11,
AND DECEMBER 9, 1909

THE CRITICISM

AN EDITORIAL IN "ENGINEERING NEWS"

NOVEMBER 11, 1909.

IN our issue of Oct. 21, 1909, we illustrated the new speed-reducing gear for marine steam turbines invented by Rear-Admiral Melville and his partner. We gave space to this device because it was attracting wide attention in the newspapers and is an ingenious attempt at working out what is unquestionably a very important problem in marine-engineering development. While we made no comment upon the device at the time of its publication, it seems proper that we should point out some of the reasons which make it extremely doubtful whether it can be successfully applied to the practical work of ship propulsion.

As our readers who have examined the device will recall, it consists of a gear and pinion with spiral teeth meshing with each other, the pinion being on the turbine shaft, of course, and the gear on the propeller shaft. In the experimental machine which the inventors have constructed, a gear tooth of 1¼ ins. pitch is used and a load is assumed upon it of about 450 lbs. per lin. in. This is claimed, doubtless with truth, to be well within the limits of safe working pressure for gears of high-quality alloy steel. But the question which will give pause to the ship owner or naval engineer who may seriously consider the introduction of such a device on board his ship, will be the question of the effect of shocks upon these gear teeth. It is perfectly well known that the forces to be met with by the propelling machinery of a ship in a rough sea defy all attempt at calculation. When the ship pitches heavily, the screw is alternately raised wholly or partially out of the water and then buried deeply in it. The force of the waves which strike the screw in a heavy sea can be appreciated to some extent by consideration of the damage done to other parts of the vessel by occasional huge breakers. It is well known that the wrenching which the propeller gets in heavy seas often causes fracture of the huge screw shafts themselves.

To introduce, now, such a delicate thing as toothed gearing, no matter of what form or of what material, between the engine shaft and the propeller shaft, is to introduce a weak link into a chain. It is manifest that the strength of these teeth cannot possibly be anything like as great as that of the shafts which they connect. As to how much pounding a gear of this sort might stand before the teeth would strip, no engineer could possibly prophesy—nothing but actual trial could determine; but in any event the weakness of the connection is so evident, and the serious danger to a vessel which should be deprived of her motive power through such a failure is so well known, that no vessel of much value is likely to be hazarded in a trial of the device.

THE REPLY

FROM *LETTERS TO THE EDITOR* "ENGINEERING NEWS"

DECEMBER 9, 1909.

SIR: I note with interest your editorial on page 519 of the issue of Engineering News for November 11, in which you express grave doubts as to the practicability of the Melville and Macalpine speed-reducing gear for marine steam turbines.

While your reasoning is perfectly sound, it is unfortunately based on unsound premises. In the first place, toothed gearing is not necessarily a delicate thing. Its wide and successful use in heavy rolling mill work in which the stresses are more sudden and severe than in marine work should, I think, be sufficient to exclude this point from further consideration.

In the second place, the assumption that it is impossible to construct a gear which shall be as strong, or even much stronger than the shaft it drives, is far from justifiable. The question as to whether the gear shall be the strongest or the weakest link in the chain, is a matter wholly within the control of the designer.

As to the possible pounding of the gears, it must be remembered that pounding cannot occur without backlash, and in accurately cut gearing the backlash is a negligible quantity.

However, as far back as history goes, every departure from established practice has been confronted with, what seemed at the time, good and sufficient reasons why it should be a failure; and if we may read any lesson at all in history, it is, that in matters pertaining to modern science and engineering, failure is becoming more and more a very unsafe prophecy.

Yours very truly,
H. E. LONGWELL,
Consulting Engineer.

East Pittsburg, Pa., November 29, 1909.

[If it is practicable to design a toothed gearing to drive the screw-propeller shaft of a large ocean steamer which will be actually stronger against sudden shocks than the shaft itself, then the criticism to which our correspondent takes exception would not hold good. The point to which we particularly desired to call attention, was the difference between the use of toothed gearing on a ship's propeller shaft and its use in ordinary machines. When a tooth breaks in the power transmission plant of a mill the mill shuts down till repairs can be made. On shipboard, however, a broken tooth on a propeller shaft might mean the loss of a vessel, or in the navy the loss of a battle.

The letter which follows, from one of the designers of this speed-reducing gear, indicates that it has been found practicable to make a toothed gear with a margin of strength over the shaft.—Ed.]

SIR: The editorial comment on the Melville and Macalpine Reduction-gear made in your issue of 11th of November was not read byme until a few days ago; but as it raises some questions of the highest importance, I am very pleased that you have given me the opportunity of discussing them publicly.

Both Admiral Melville and I are marine engineers of long experience, fully acquainted with conditions at sea. Therefore, the possible damage from the propeller meeting an obstruction such as a log, or the effect of rough weather with the consequent racing of the propeller, naturally were given very full consideration.

In the design of the Melville and Macalpine Reduction-gear, which has been built and so successfully tested by Mr. Geo. Westinghouse in the works of The Westinghouse Machine Company, the pitch circle of the large gear is 70 inches diameter and the hollow shaft carrying this gear is 11 inches outside diameter and 6 inches inside. Therefore the tooth forces are exerted at over six times the radius of the shaft. It is manifest then that the teeth are very advantageously placed for stressing the shaft.

Whether we regard the area to be sheared when the teeth fail, or look on them as very short and relatively deep beams submitted to crossbending, calculation shows that they are several times as strong as the shaft. In judging of this strength, it should be remembered that the whole object of the design is to produce continuous line contact along the teeth, under all practical conditions, and the experiments show that this has been accomplished very perfectly. This perfect tooth contact greatly increases the strength over that, for instance, of railroad motor gears which are so subject to errors of alignment, giving rise to point contact of the teeth; in which, consequently, only small parts of the teeth are contributing to the strength at any instant.

But let us apply the ordinary crossbending formula on the supposition that *all* the forces between the teeth are exerted at the points of the teeth—a condition far in excess of the truth and quite impossible in these helical gears. When the gear is transmitting 6000 horse-power and the pinion running at 1500 revolutions per minute, the pressure at the tooth contacts is 450 pounds per lineal inch. This applied at the point of the teeth would, by ordinary crossbending formula, which gives an excessive value of the stresses for such short beams, induce a maximum fibre tension of 4500 pounds per square inch. Quite a common commercial steel is used of 70,000 pounds tensile strength and 35,000 pounds elastic limit. Thus, from the ultimate strength we have a factor of safety of 15.5 and, from the elastic limit, of 7.8, even with the greatly exaggerated conditions. This is an ample margin for all contingencies and probably the actual factors of safety are double these just given. At this power and speed, the torque on the shaft would induce a shearing stress of 5,160 pounds per square inch. With this must be combined the considerable stresses due to crossbending of the shaft. Remembering that the shearing stress of the steel is probably not more than 72 per cent. of its tensile strength, we see that, even with the unfavorable supposition, the teeth are about 60 per cent. stronger than the shaft; and actually, no doubt, two or three times as strong. It should be noted that the shaft is of ample size for a crank shaft transmitting the same torque, in which case it would have much heavier crossbending stresses than in the case of the gear.

Again, in thinking of the propeller striking a log or the water, the flywheel action of the large gear and propeller should be carefully kept in mind. This strongly protects the teeth from shock. Compare the case of a rolling mill engine driving rolls through gears, where the load of the engine rises from that due to friction, to full or over-load instantly. Even in this extreme case the shock, which would otherwise be disastrous to the gear, is effectively softened by the energy of the flywheel placed between the gears and the rolls.

Again, the rise of stress to which the teeth are subjected depends not only on the amount of retardation of the heavy flywheel formed by the propeller, shafting, and large gear, but also on the moment of inertia of the turbine and pinion, which are again very elastically connected. If these parts were massless, the teeth could receive no shock.

In the tests, the pinion is driven by a double flow turbine running usually at 1500 r.p.m. and developing up to 6700 horse-power. Suppose a retardation so excessive that it would diminish the speed of rotation of the turbine 10 per cent., or 150 r.p.m., in one second; in other words, such as would stop the turbine in 10 seconds if continuously applied. Calculation from the actual moment of inertia of the turbine rotor and pinion shows that at 6000 horse-power and 1500 r.p.m., the steam not being throttled down, the consequent rise of stress in the teeth would only be 33 per cent.

This supposes a very excessive case. Such severe racing would not be allowed. The ship would be run at reduced power and the tooth stress would probably never approach, in such weather, the smooth water stress due to 6000 horse-power.

Further, the high speed turbine allows of a very sensitive governor which, when the propeller begins to race, would at once reduce the power so that when the propeller is resubmerged the steam torque is much below the maximum.

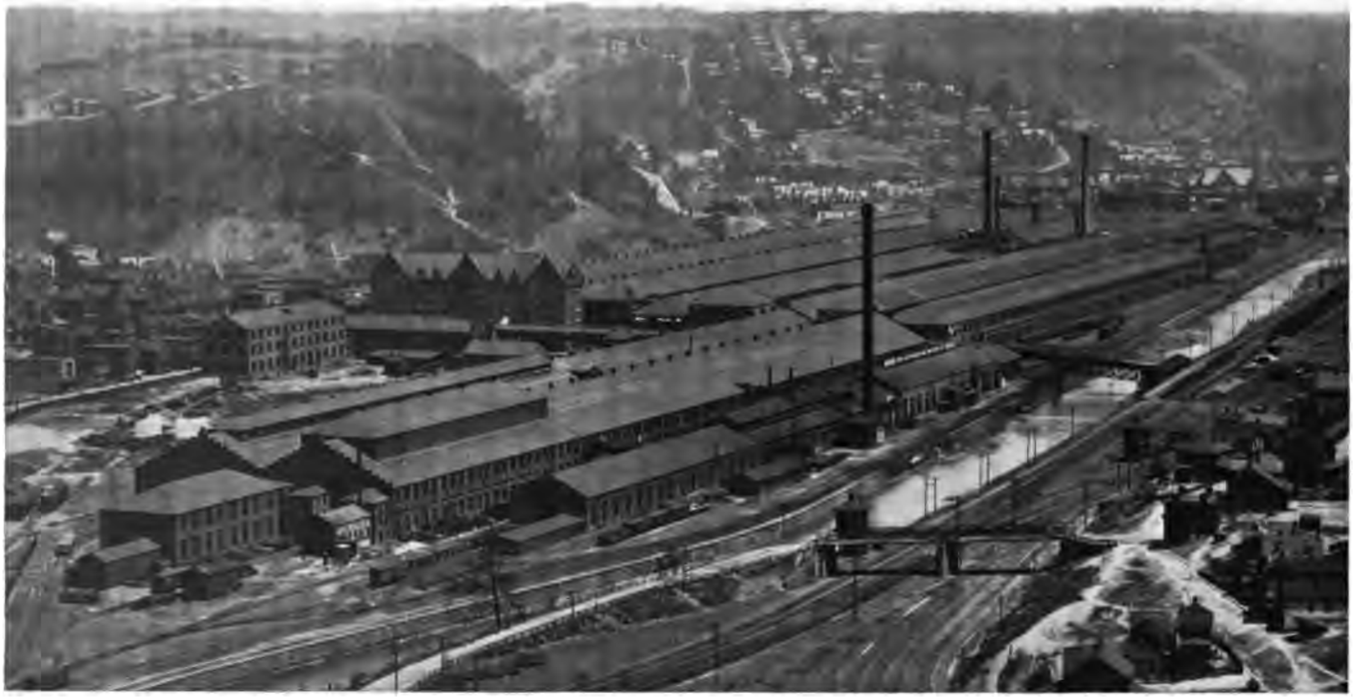
Lastly, we have practically no wear, as the tool marks still show prominently in the teeth. This is partly due to copious lubrication and partly to the well distributed tooth contact. Hence the life of the pinion will be long, possibly as long as that of the ship. The perfect action of the whole gear is shown by the high efficiency attained which, at 6000 horse-power, is over 98.5% and probably as high as 99%.

By using a steel of high elastic limit, but still ductile, we have no doubt we could with perfect safety raise the power transmitted by the present machine, which weighs 61,000 pounds, to 10,000 horse-power at 1500 r.p.m.; and there is no indication that this is the limit of peripheral speed at which these gears can be run.

Yours faithfully,

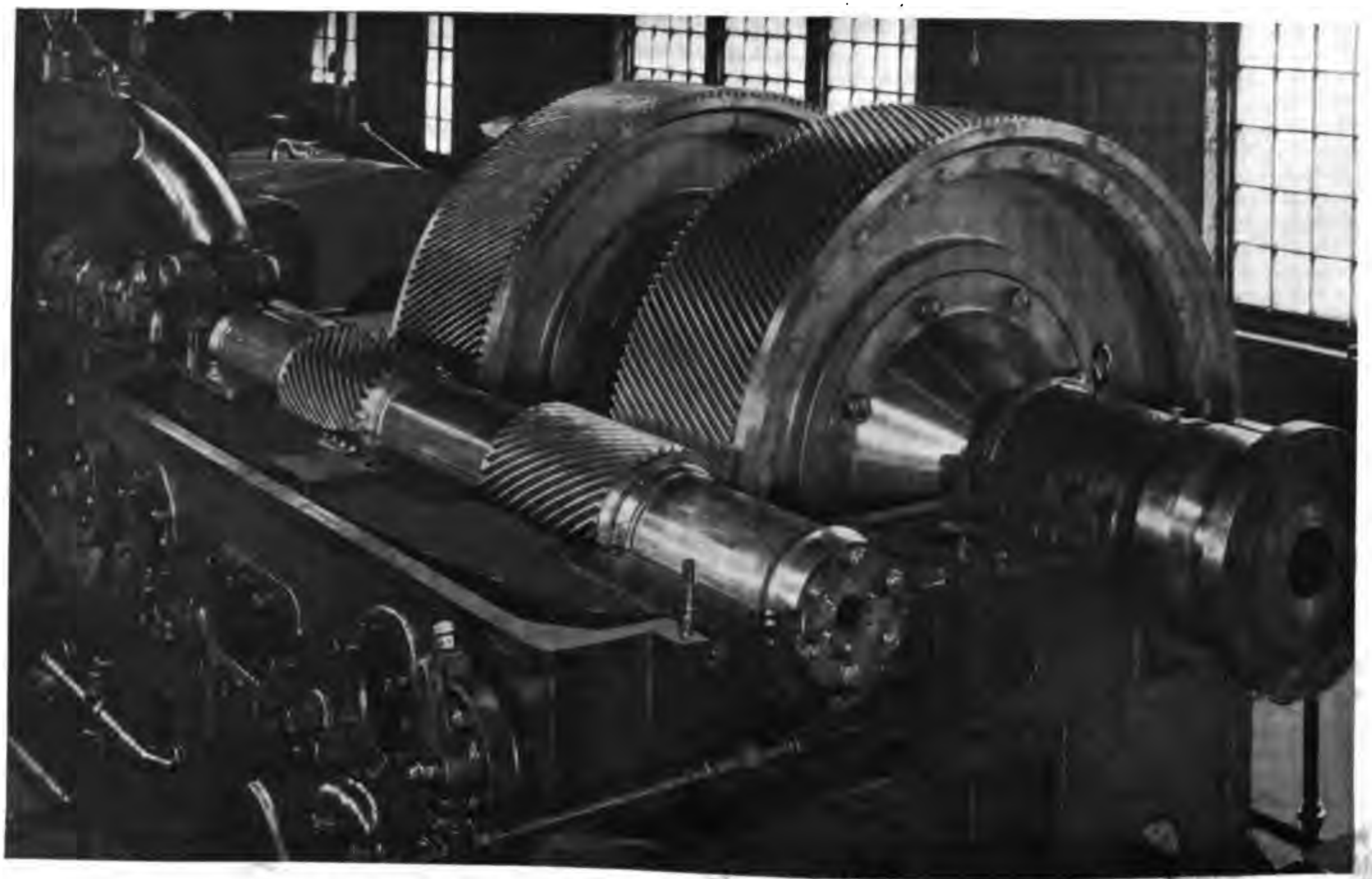
John H. Macalpine.

532 Walnut St., Philadelphia,
November 29, 1909.

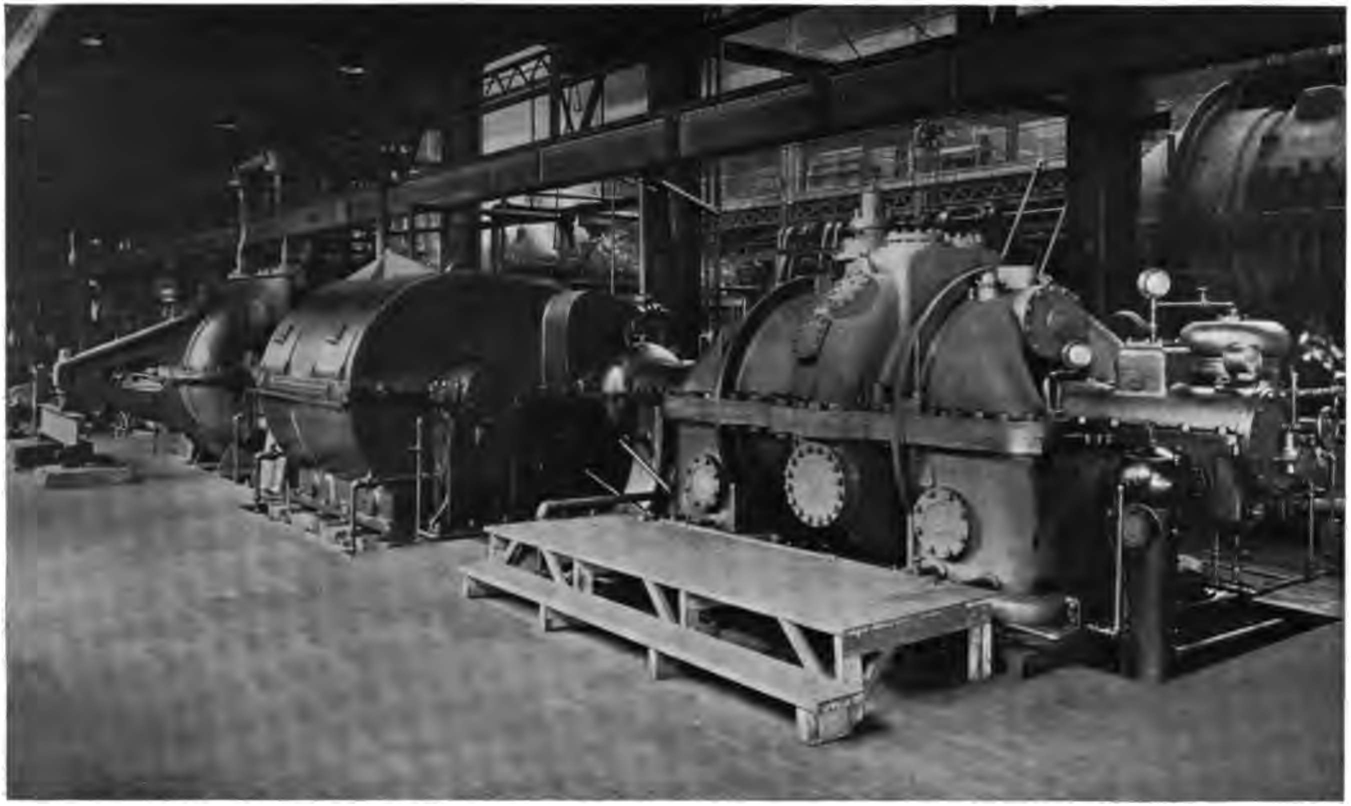


The Westinghouse Machine Co. Westinghouse Electric & Mfg. Co.

WESTINGHOUSE WORKS AT EAST PITTSBURG, PA.



VIEW OF GEAR AND PINION—COVER AND UPPER HALF OF FLOATING FRAME REMOVED.



EXPERIMENTAL INSTALLATION AT THE WESTINGHOUSE MACHINE COMPANY'S WORKS, 6000 H. P. TURBINE, REDUCTION GEAR AND DYNAMOMETER

APPENDIX

A REPRINT OF A
REPORT ON STEAM TURBINES
SUBMITTED BY MELVILLE
AND MACALPINE MAY, 1904

HERO'S TURBINE

REPORT ON STEAM TURBINES

BY

MELVILLE & MACALPINE
OF PHILADELPHIA

FOR

GEORGE WESTINGHOUSE

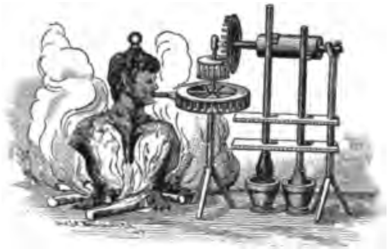
SUBMITTED MAY 1904

Copyright, 1904, by
George Westinghouse
Pittsburg, Pa.

INTRODUCTORY NOTE

THE following report was made for me by Rear-Admiral George W. Melville, ex-Engineer-in-chief, U. S. Navy, and Mr. John H. Macalpine, Consulting Engineers. As I consider the information gained and views expressed of value, I have had printed a limited number of copies for private circulation in order that others might benefit. As it was a purely private report made for my own use, names were quite freely mentioned and opinions given in conversation quoted—it being known, however, that a special investigation was being made. I hope it will not be felt by anyone that an indiscretion has been committed in distributing these copies.

Geo. Westinghouse



BRANCA'S TURBINE

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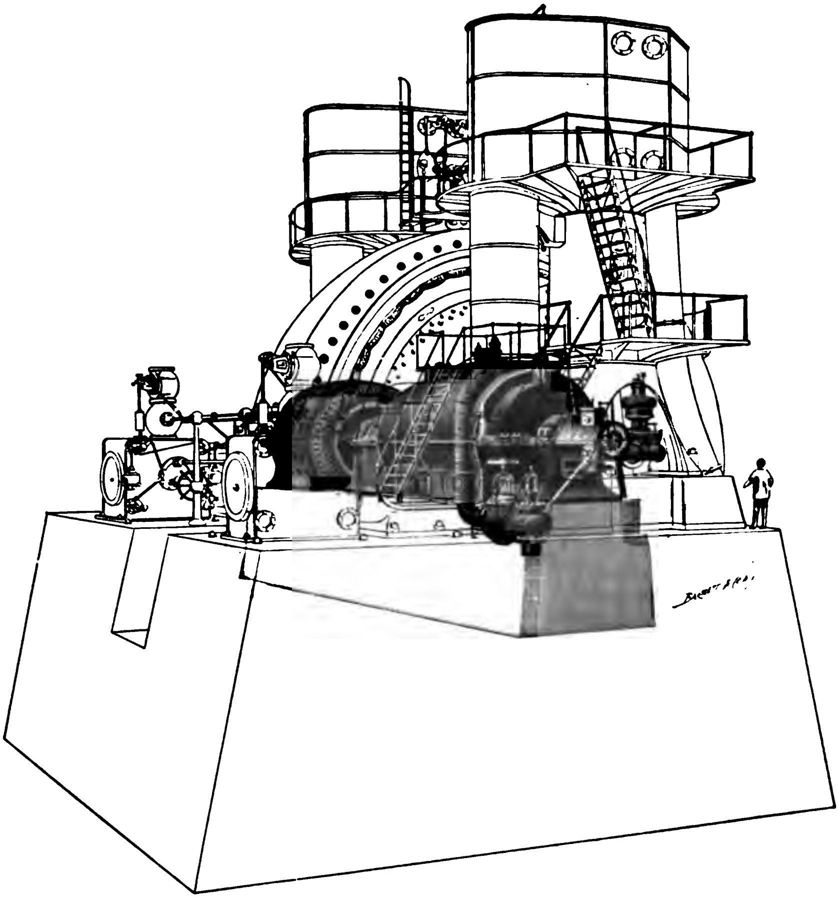
REPORT ON STEAM TURBINES

BEING PRINCIPALLY THE RESULTS OF AN INVESTIGATION IN EUROPE,
ESPECIALLY WITH REGARD TO THE APPLICATION OF THE
TURBINE TO MARINE PURPOSES.



COMPARISON OF SPACE OCCUPIED BY AND FOUNDATION NECESSARY FOR A 5000 KILOWATT RECIPROCATING ENGINE GENERATOR UNIT AND A WESTINGHOUSE-PARSONS TURBINE-GENERATOR UNIT OF A SIMILAR RATING AND OVERLOAD CAPACITY

INTRODUCTION

THE MARINE TURBINE has been the object of very great professional and popular attention within the last few years and much has been written on the subject. The claims made for it have varied in their extent, but usually have been very large. To any who have been strongly influenced by this literature the following Report, in which we have tried impartially to state the truth so far as we know it, must appear antagonistic.

Much is set down here which was given to us in the course of conversation, but as those we were speaking to knew, we believe in every case, that we were investigating the status of the marine turbine, we do not think we have overstepped the bounds of propriety in what we have written. It is hardly to be expected that all through we have given with perfect accuracy the opinions of our informants and in other ways errors may have crept in, though we have striven to be correct. Our own conclusions are based partly on the broad consensus of opinion expressed to us, giving the experience and beliefs of many leading men in our profession, and partly on our own independent knowledge. Any slight error or unintentional misrepresentation would not sensibly change the color of this Report.

We believe that the marine turbine is still at the experimental stage, but that possibly it may have a considerable field in the future. How broad that field will be is still far from settled, but that the reciprocating engine will retain much the larger field we feel no doubt at all.

To refer to the reciprocating engine as an "unscientific battledore and shuttlecock arrangement"—as was done by a gallant Admiral last year, in discussing a paper by the Hon. Chas. A. Parsons, at a meeting of the Naval Architects — is utterly reckless and foolish. Surely it is not open to contention that the reciprocating engine has played its role admirably and has aided civilization immensely more than any other machine ever invented by man. As it stands today, in its many forms, it is the product of the brains and hands of a legion of our most skillful and scientific mechanics. In its high economy over a very extended range of power and its ability to reverse, it possesses two of the most essential qualities in a motor,— qualities which the turbine entirely lacks and which we believe it always will lack.

A rotary motor has naturally always appeared to engineers as an ideal form of engine and the turbine is now, without doubt, admirably suited to some applications where high speed of rotation is an advantage. But among many of the advocates of the turbine there has, we think, been a tendency — more or less unconscious, no doubt, and arising from very laudable enthusiasm in a worthy cause — to disparage the splendid qualities of the reciprocating engine and far overstate the relative advantages and economy of the turbine even in its most favorable land applications.

This is very unfortunate, as progress is more readily assured by dispassionately recognizing the present position and the difficulties to be met.

How the turbine's range of efficiency is to be extended to lower peripheral speeds without great addition of weight, we have not been able to conceive, but if this is not realized its application will be confined to very fast ships which have to run slow a very slight proportion of their time.

Before proceeding with our Report, we offer our best thanks to Messrs. A. F. Yarrow, Wm. Denny & Brothers, The Allgemeine Electricitäts-Gesellschaft, Escher Wyss & Co., Sautter Harlé & Co., and Munn & Co., proprietors of the Scientific American, to whom we applied for illustrations when it was determined by Mr. Westinghouse to print this Report. These all add great interest to the text.

We should also state here that all the turbine steamers shown by these illustrations were, we believe, engined by the company of which the Hon. Chas. A. Parsons is the head.

CLAIMS FOR THE TURBINE

The claims put forward for the marine turbine as compared with the ordinary reciprocating engine, by the Hon. Charles A. Parsons, and by others usually quoting from him, may be summed up as follows,—

1. Reduced weight per horse-power, (a) of machinery, (b) of keelsons;
2. Reduced space required for machinery;
3. Increased economy, (a) of steam, (b) of attendance, (c) of oil and other stores;
4. Perfect balancing of the turbine;
5. Reduction of propeller diameter;
6. Reduction of initial cost;
7. Reduction of cost of up-keep and overhaul.

If these claims were substantiated, the following advantages would be attained,—

1st. From reduced weight per horse-power there follows a reduction of displacement and consequently of resistance; which, in turn, means reduction of the power required for a ship doing the same duty. That is, the reduction of weight of machinery and displacement react on one another, giving a final advantage which is considerably greater than in direct proportion to the original reduced weight per horse-power. The amount of the final gain would depend on what proportion of the total displacement is due to the machinery. This reaches a maximum of about fifty per cent. in torpedo boats and a not much smaller percentage in some of the Atlantic Greyhounds. Hence, for all fast steamers a reduction of weight per horse-power is very valuable. The smaller ships that would then be required for the same duty involve less first cost and reduced running expenses.

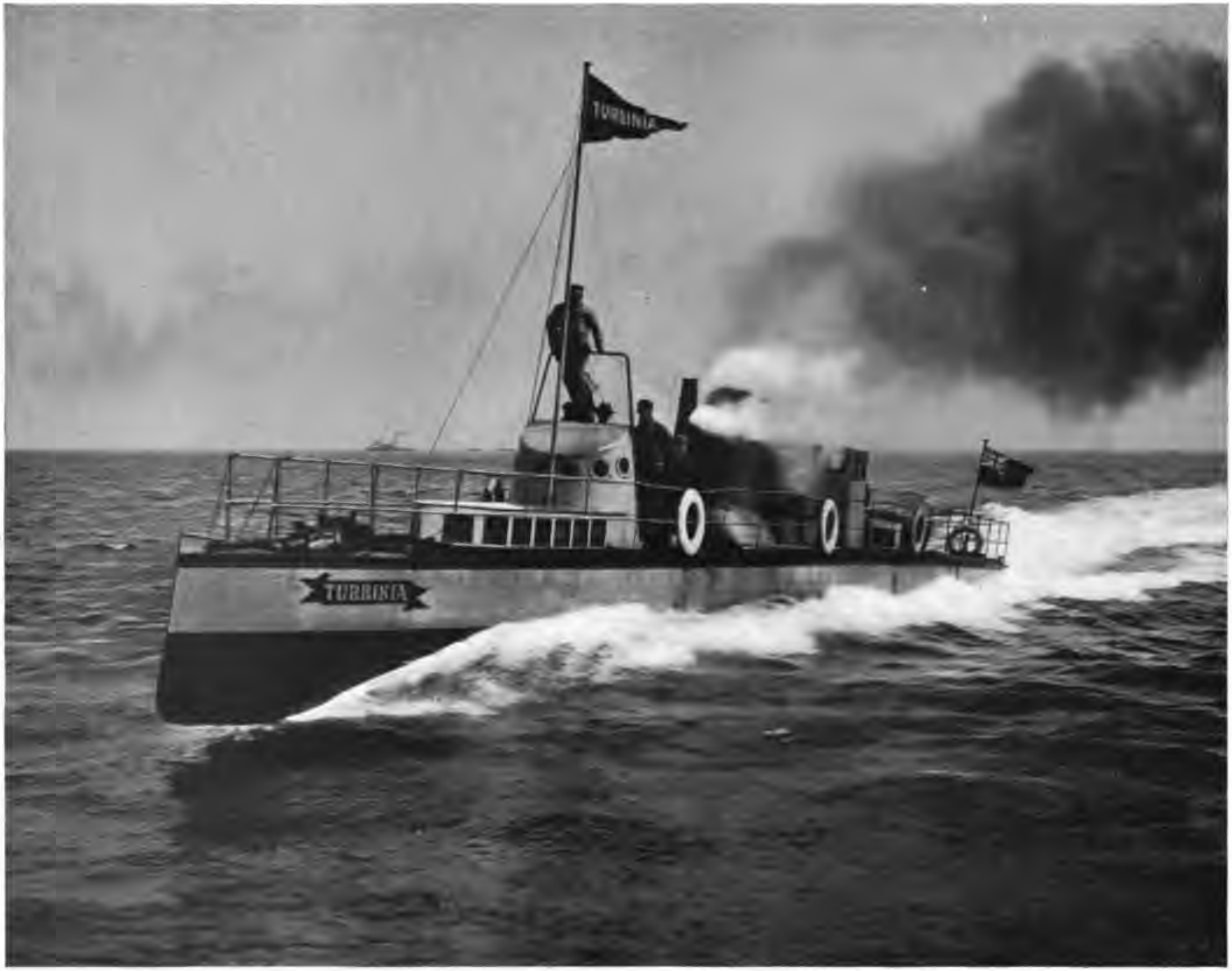
2d. Reduction of machinery space gives more to other accommodations. The turbines not requiring so much head room as reciprocating engines, the decks above the machinery level can be more fully utilized if this is thought judicious. Such low-lying machinery would also be safer in war vessels as it is more completely under water, and the lower center of gravity gives increased stability to the vessel.

3d. Increased economy of steam would lead to reduction of weight and size both in the boilers and some of the principal auxiliaries, and would diminish the bunker coal required, and the running expenses.

The reduction of machinery weight and displacement would again react on one another as under the first head.

This saving, if established, would be the most important gain of all.

Reduction of costs for attendance and stores go directly to increase the owner's profit or to reduce the cost of freight or passage.



THE TURBINIA STEAMING 35 KNOTS

4th. Perfect balance of the turbines would obviate vibration being produced by them, and greatly increase the comfort of the passengers. Excessive vibration, on the other hand, undoubtedly strains the ship.

It has been claimed that the absence of reciprocating parts, by which balance is attained, increases the safety of the crew and ship.

5th. Reduction of the propeller diameter would allow of greater minimum immersion of the blade tips (when the blade is at its top center) and consequently there would be less liability to race in rough weather.

Or the reduction of diameter may be utilized to give diminished maximum immersion of the blade tips (when the blade is at its bottom center) and thus facilitate navigation in shallow waters.

These two advantages are obviously somewhat antagonistic.

6th. The advantages of reduction of initial cost are manifest.

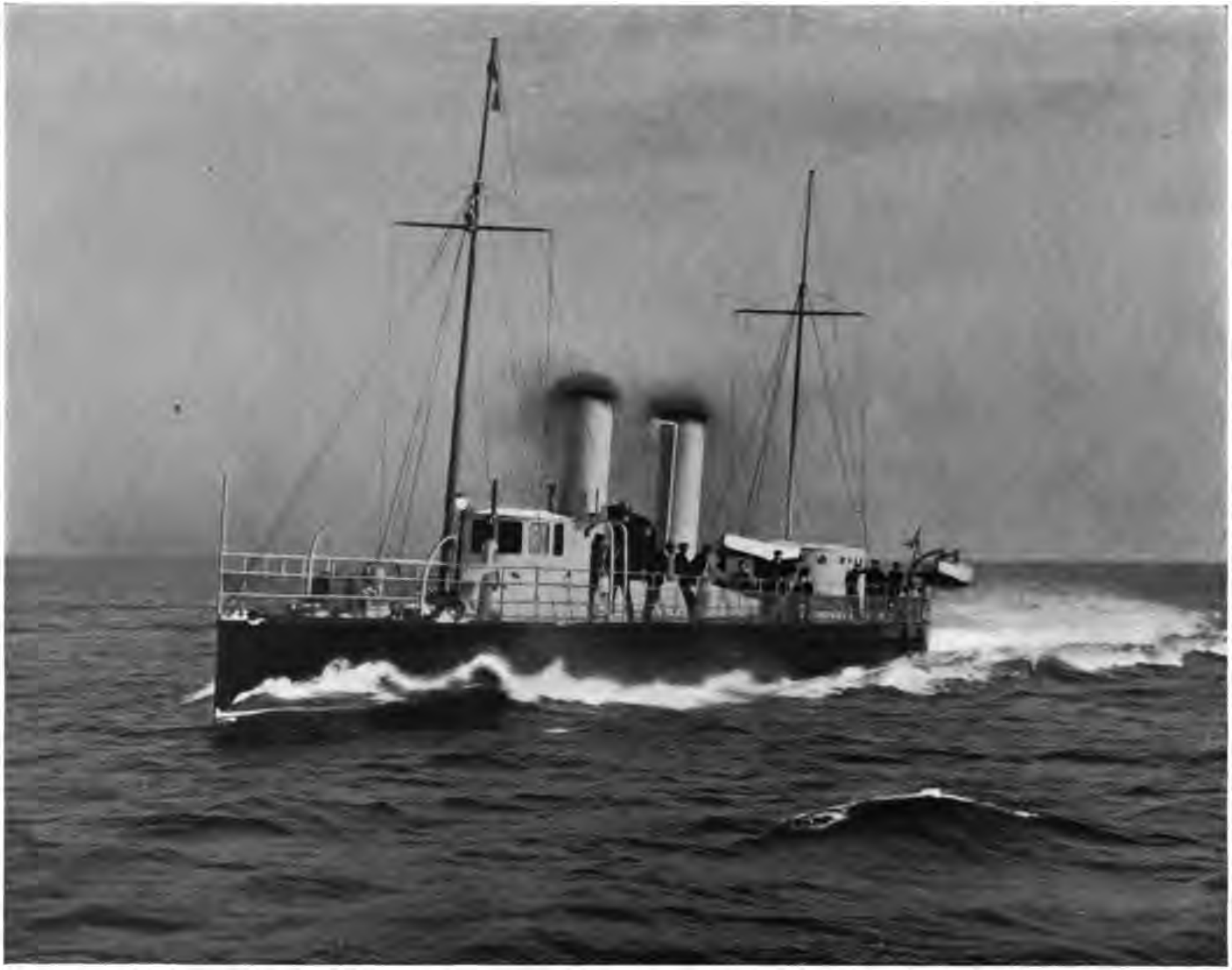
7th. Reduction of cost of upkeep and overhaul would not only give similar advantages to those of the reduction of initial cost, but it also means that the ships would be laid up and unremunerative a smaller proportion of the time they are in service.

The examination of these claims naturally determined the principal course of our investigation, and we will discuss them before giving other results and data which we gathered or describing some of the turbines that came under our observation

DISCUSSION OF CLAIMS

I.—WEIGHT

Referring to the "King Edward," Mr. James Denny made the statement that if fitted with twin triple-expansion engines the best speed that could have been attained with the same boiler was 19.7 knots as against 20.48 knots actually attained by the turbine. This is the most definite statement we have seen, and if borne out by subsequent experience would constitute a noteworthy advance. That she should be an advance in this respect on the paddle engines used in practically all the passenger steamers of the Clyde, was to be expected. These held their position purely on account of their quick stopping and starting, notwithstanding their low piston speed and the inefficiency of the paddle, which leads to great additional weight in the machinery. And, except for the "Queen Alexandra," the paddle is still adopted in all new passenger steamers on that estuary, notwithstanding the experience with these two boats. The information we obtained does not seem to support Mr. Denny's claim. But on our visit to Dumbarton, Mr. Leslie Denny said that undoubtedly they had a saving of weight and, as an instance, gave the weight of the keelsons, or engine seating, for one



ENGLISH TURBINE YACHT "TARANTULA," BUILT BY MESSRS. YARROW & CO.

of their ships as about 2 tons against 20 tons, which would have been required had piston engines been adopted.

As showing how excessive are the weights for paddle engines, we give a figure quoted to us by Mr. Bruce Cumming, Superintendent of the Larne and Stranraer Line of Steamers, for which the Dennys have just finished the "Princess Maud," of 6,000 IHP and 20 knots,—

Turbines and shafts for the "Princess Maud" weigh about 60 tons, while a corresponding paddle steamer had, for the paddles and shafts alone, 103 tons.

*Mr. A. F. Yarrow has built several torpedo boats which are sister ships, except that some have had reciprocating engines, one Parsons turbines, and one Rateau turbines. These are identical in lines, hull construction, boilers, etc. He definitely stated he had effected no saving of weight by adopting the turbine. This seems the more remarkable, as we would have anticipated some sensible advantage in this respect in the Rateau over the Parsons turbine.

In our various visits to the Admiralty, in London, and at other times, we met Admiral May, Mr. Philip Watts, Sir John Durston, and Admiral Oram. Speaking of the "Velox" and "Eden," which have turbines, compared with the sister ships which have reciprocating engines, the uniform and emphatic testimony was that there was *no* saving of weight by adopting the turbine.

This was certainly true of the "Cobra," the weights of whose machinery were given at the inquiry into her loss. The following instructive quotation is from an editorial on "The 'Cobra' Disaster," in Engineering, Vol. lxxii, p. 553,—

"She was, however, mainly experimental, and was a larger and more highly-powered vessel than any of her class, which had up to then been added to the Navy, her length being no less than 223 ft. 6 in. Her machinery, too, came out 30 tons heavier than was anticipated, the total weight being 183 tons. These figures may be compared with those of two former destroyers built at Elswick for the Royal Navy—the 'Swordfish' and the 'Spitfire'—which were 200 ft. long, and had machinery weighing 110 tons. It is worth noting that the extra weight of the 'Cobra's' machinery was estimated to give her 60 per cent. more power than there was in the 'Swordfish.'"

Accepting as accurate the estimate of increased power over that of the "Swordfish," and taking the weight per horse-power the same for the reciprocating engine over this range of increase—which is not far from the truth—the "Swordfish" with 60 per cent. extra power would have had machinery weighing 176 tons, and the weight of the turbine machinery comes out 4 per cent. heavier per horse-power. As the turbine was the novel part of the outfit, it is practically certain that the large overweight occurred in connection with it and its accessories and that its weight very considerably exceeded that of the corresponding reciprocating engine which it replaced.

---

*It may be noted in passing that Mr. Yarrow is building another sister ship to the above with a turbine of a design he is making along with Mr. Fullagar. The nature of this design was not disclosed, but Mr. Yarrow based considerable hope on it.

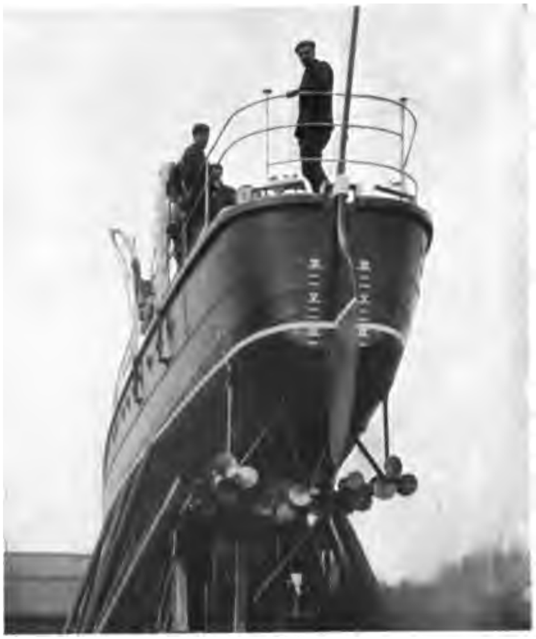
The same error of computation seems to have occurred in the "Eden"—see Table III page 25 and the explanation immediately following that table—though in this case it seems to have affected the boats with ordinary machinery also.

Again, we have it on what we consider good authority that the Turbine yacht "Tarantula" was largely a failure because the turbines were "so heavy that she trims very badly by the stern——."

Mr. Parsons might object that torpedo boats were a somewhat unfavorable type for a comparison of weight, as he states that in this respect the advantage of turbines increases with higher powers and speeds. His first claim at the end of his paper on "The Marine Steam Turbine and Its Application to Fast Vessels," read before the Institution of Engineers and Shipbuilders in Scotland, is,—

"(1) Increased speed for the same boiler power, due to considerably reduced weight of machinery, and increased economy in steam. (This advantage increasing with higher powers and speeds.)"

The vague statement in parenthesis seems, from the portion of Mr. Parsons' paper preceding that quoted, to refer to a comparison of large and small fast vessels of the same type. Also the claim for reduction of weight is made separately from that for the economy of steam (with which we deal later) and seems to refer to a supposed increasing advantage in weight per horse-power as the power rises. We do not think such a belief is well founded. If two similar engines, with the same piston speed if reciprocating, or the same peripheral velocity if turbines, and working at the same pressure, be compared, then, omitting gravity stresses, the metal in both cases will be stressed in corresponding parts with equal intensity in the large and small machines; that is, the mechanical design, if correct for one, will be correct for the other. A moment's thought will show that the weight per horse-power will therefore be proportional to the linear dimensions. Much the same economies in weight from that of a strictly



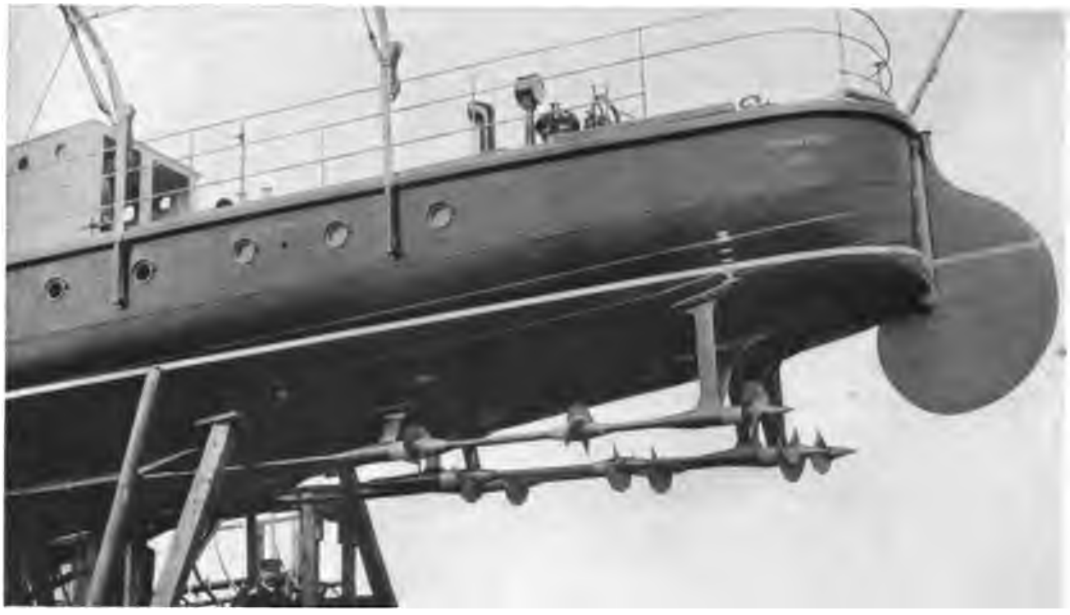
THE "TARANTULA"
STERN VIEW SHOWING THE NINE PROPELLERS

similar design can be made in the reciprocating engine as in the turbine. For instance, in enlarging the turbine one cannot, except within moderate limits, increase the length of a blade without also increasing its breadth and thickness and thus increasing the turbine length. Hence the above rule for weight per horse-power will obtain with fair accuracy.

This contention seems to be borne out by the rumor that the anticipated saving of weight in machinery in the new 25,000-ton Cunarders is 300 tons. As the total weight of machinery will be in the neighborhood of 12,000 tons, this saving of 2½ per cent. is well within the usual excess of actual over anticipated weights; and, taken in conjunction with the rumored anticipation, by Mr. Parsons, of an increase of 5 per cent. in economy, the claimed advantage in weight per horse-power in large-sized turbines seems more than to vanish.

From the evidence before us we do not think that any claim for more than very slight, if indeed any, saving of weight by adopting the turbine can be maintained. We would anticipate that such turbines as the Rateau or Zoelly would have some advantage over the Parsons in this respect, but we would judge from Mr. Yarrow's statement, noted above, that the gain is a vanishing quantity.

Until there is a decided gain in economy by use of the turbine, allowing a reduction of boiler weights, etc., no important saving of weight can possibly result,



VIEW OF PROPELLERS, SHAFTING, AND BRACKETS OF "TARANTULA" for the main engines form a comparatively small fraction of the whole machinery weights.

II.—SPACE

Admiral May said there is a saving of machinery space by adopting the turbine, while Mr. Yarrow said there is not. From all the evidence we gathered, the truth seems to be that there is little or no saving of floor area, but that there is a considerable saving of space from the reduced head room required. This seems to reconcile the above contradictory statements, as it is not in all respects desirable to save head room, and probably Mr. Yarrow did not do so.

The increased safety in warships due to the turbines lying low is an unquestionable advantage. If, however, the engine room is almost completely ceiled by a low-lying deck, as seems to have been done in the turbine ships of the British Navy, we return to the same condition as for horizontal engines, and the staff have to work with their heads in foul air. This objection was very strongly stated to us by Sir John Durston as obtaining in the turbine ships in his charge. It cannot be urged, however, as any objection to the turbine; as it is quite open to the designer to judge what degree of overhead freedom gives the best advantage.

In passenger and merchant ships there could be no question that the space overhead should be left very open, and this is certainly done on the turbine steamers "King Edward" and "Queen." Besides, as the Hon. W. J. Pirrie pointed out, for all steamers coming under Board of Trade rules, there is no object in curtailing the machinery space. The port dues, etc., are based on register tonnage. If, in a screw steamer, the propelling space is not less than 13 per cent. of the gross tonnage, a reduction of 32 per cent. of the gross tonnage is allowed; and if the percentage for the propelling space is above 20 per cent. even greater inducements are held out to the owner. Thus, the reduction of machinery space, in a large proportion of steamers, would mean greatly increased running costs.

But even when this Board of Trade regulation has no effect, we think it would be quite inadvisable in ordinary steamers to curtail the head room and thus make the engine room stuffy and dark; and for such we consider the claim of diminished machinery space, since it is unaccompanied by sensibly diminished floor space, of little value.

Possibly in warships the reasonable comfort of the staff, which is an essential element in determining fighting chances, could be sufficiently attended to and still the increased defensive power of a low deck in the engine room retained.

The increased stability which has been claimed from the low-lying machinery must be of very minute amount, and as ample stability can readily be obtained by good design we do not think any weight should be attached to it. It should be remembered that only a limited metacentric height is desirable. For instance, if the metacentric height of the "Cedric" were increased, her natural period of roll would be diminished and she would then respond more readily to a beam sea. At present her rolling period is too long to respond and thereby the comfort of the passengers is greatly increased. A vessel with an excessive metacentric height is very wearing to the passengers and crew, as the writers have had reason to know.

III (a).—ECONOMY OF STEAM

This is the most important consideration of all and we devoted very special attention to it.

On land, in electrical work for instance, the turbine can be driven fast and economy is gained by high peripheral speed. In marine practice the propeller will be efficient only when driven at a comparatively slow speed so as not to use a very small pitch ratio. Hence either, first, the peripheral speed must be reduced; or, second, the diameter of the turbine made forbiddingly large; or, third, the number of stages or blade rings must be increased. In any case we cannot usually look for the same steam economy as on land. The propeller does not impose the same restrictions on the reciprocating engine.

The chief engineer of the "Queen" informed us that she burned, on an average, twenty tons of coal in the twenty-four hours. Steam was kept up for shifting the boat at the dock; but, of course, by far the greatest consumption was during the two hours occupied by the double journey. The slower paddle boats of the same fleet, he said, burned about fourteen tons per day. We had hoped, in order to analyse this result, to get information regarding the size and power of these other boats before this report was completed; unfortunately, we have been disappointed. But the "Queen" occupies one hour in crossing on the average, or a little over, while the other boats take about one hour and ten minutes; that is, their speeds are roughly as six to seven. For exactly the same hulls and economy of performance, the coal burned would be as the resistances,—as the squares of the speed—since the distance traversed is the same for all the boats.

$$6^2 : 7^2 :: 14 : 19.1 ;$$

therefore the figures taken, if anywhere near right, do not indicate much economy in the "Queen" unless the paddle boats are very much smaller. They are far from indicating any economy when compared with screw engines, which may be taken as 10 to 20 per cent. more economical; though, curiously, Mr. Barnaby states in "Marine Propellers" that the paddle is not inferior.

This comparison, we are aware, does not carry very much weight on account of the possible inaccuracy of the figures. But, it must be recollected that the Chief

Engineer coals his boat nearly every day and the quantity required for his own boat is sure to be a figure well known to him; while that for the other boats of the fleet is also likely to be known definitely in a case like this, where a new departure in the machinery has been introduced and the main interest to the company was this one of economy. Also, the coal burned while running would be approximately as the coal per day. The figures were given us quite frankly and we have no reason to suspect that we were being misled. If any misrepresentation had been intended the figures given would hardly have left the economy of the turbine so much in doubt.

At the Continental Department of the London, Brighton, and South Coast Railway Company, London Bridge, the manager told us that they were by no means sure whether their turbine steamer "Brighton" was more or less economical than their boats with reciprocating engines. He said, the Cunard Commission has made comparative tests of the "Brighton" and others, but had kept the data secret even from the Railway Company. Evidently the increased economy, if it existed, was not of startling amount as the coal account would then surely have shown it over the six months the "Brighton" had been running.

Mr. Leslie Denny seemed to have no doubt that in the turbine vessels they had built they had economy of fuel, though he brought forward no figures. (Presumably the comparison in his mind was with paddle steamers.) But when asked regarding the seven ships they had in their yard at the time of our visit, he said they had not in any case cut down the size of boilers from what they would have been with ordinary engines, which seems to show very slender belief in economy.

If the turbine is to take a large place in marine propulsion it must *easily* beat the paddle engine in economy; for, partly on account of its low piston speed, which increases the losses due to condensation, leakage, etc., and partly on account of the inferior efficiency of the paddle to the screw as a propeller, the paddle engine is not a very economical machine. The determining comparison must be with screw engines, to which we now come.

FIRST-CLASS TORPEDO BOAT, BUILT BY MESSRS. YARROW & CO.

At the Continental Department of the South Eastern and Chatham Railway Company, London Bridge Station, we were told that the "Queen" was economical in steam compared with the "Arundel," a recent twin-screw steamer of almost the same dimensions as the "Queen." Both ships were built by Messrs. Denny Brothers, at Dumbarton. The statement seemed to be founded on a general impression, as they could give no figures.

The Hon. W. J. Pirrie told us that he offered the owners of the Turbine Yacht "Revolution" a substantial sum to let him make a test of her economy, but that they absolutely declined. This vessel is used for exploiting the Curtis turbine, and if the economy had been good, this would have been a most excellent advertisement. We understand that the principal trouble of this boat is that the boiler power, though apparently ample in ordinary circumstances, is not nearly large enough for her.

Small high speed war vessels should, it seems to us, give a basis for test quite as favorable to the turbine as any other class, if not more so.

Mr. Yarrow's evidence is that there is no saving of coal over the reciprocating engine, even at full speed.

Precisely the same was told to us by all the before-mentioned officials of the British Admiralty.

We now come to the most serious drawback of the turbine.

It will be noted that in his publications, Mr. Parsons has always exploited the marine turbine for "fast vessels." In tramps, for instance, if the turbine was run so fast as to give reasonable economy per brake horse-power, the propeller would have such a small pitch ratio that its efficiency would be forbiddingly low and the total economy of propulsion would be far below an allowable limit. But unfortunately this excessive waste of coal also appears in a fast vessel when it is run slow.

This has necessarily been the uniform experience, and it was so stated by all we spoke with who had experience with marine turbines. We need only give the figures obtained at the British Admiralty.

Tables I and II were handed to us by the Admiralty, but it was thought that the first line of Table II, referring to the "Eden," may not be quite fair to Mr. Parsons, as he might be able to improve on the result given, and was experimenting in this direction.*

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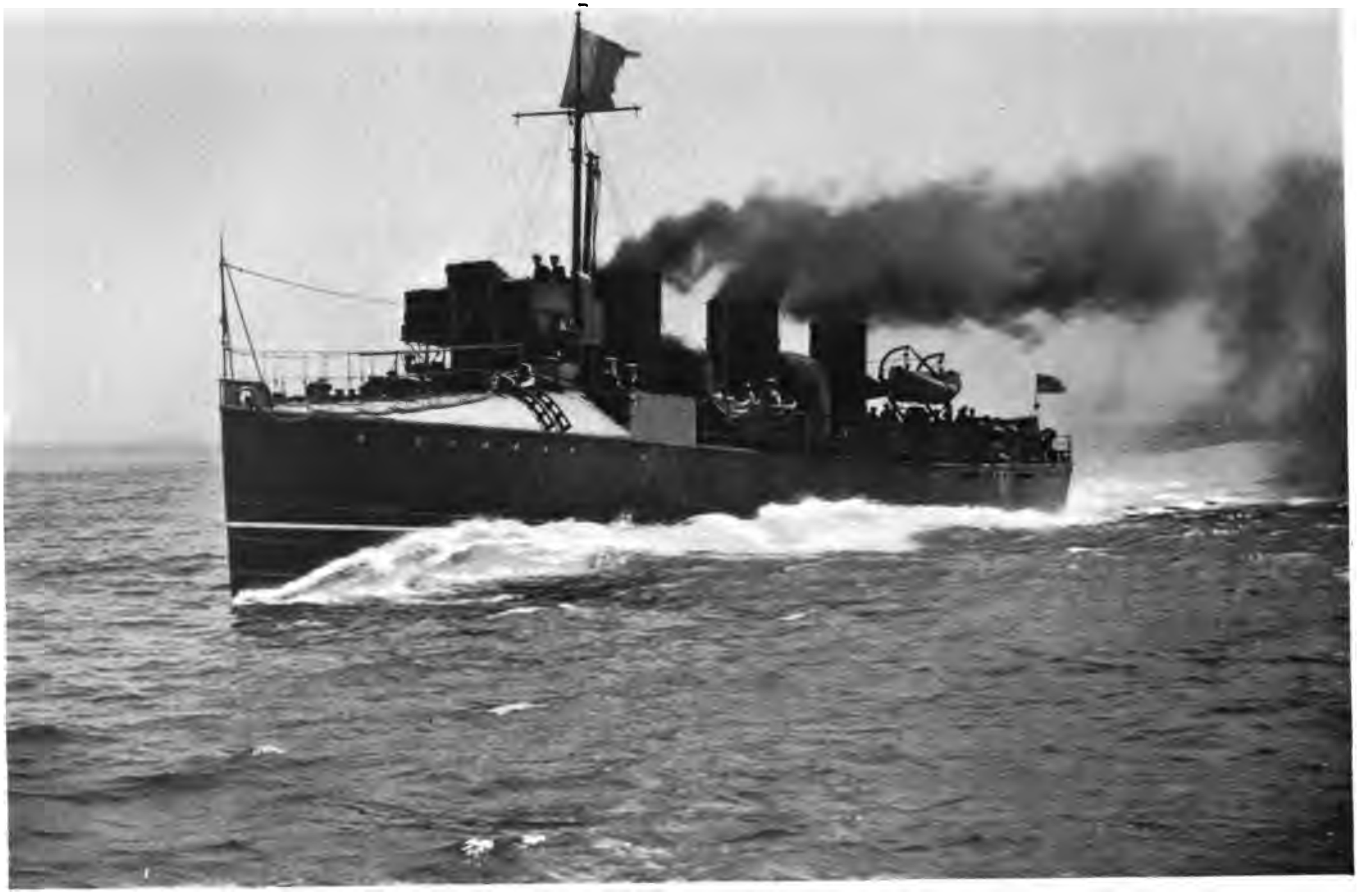
*(Added Oct. 19, 1904.) We have heard that a considerably improved result at half speed was obtained with the "Eden" in trials subsequent to our visits to the Admiralty, but that she had still nearly twice the consumption of the ships with reciprocating engines. Thus there is no reason to modify the opinions we express.



THE "VIPER," BUILT AT ELSWICK

TABLE I.

FULL SPEED COAL CONSUMPTION TRIALS.

Name of Ship.	Duration of Trial in Hours.	Speed in Knots.	Water Lost in Tons.	I. H. P.	Total Coal Used, in Tons.	Kind of Engine.
Eden . . .	4	25.287	6.0	......	29.825	Turbine.
Exe . . .	4	25.292	2.25	6856	25.8	Reciprocating.
Itchen . . .	4	25.250	4.08	6805	29.9	do.
Usk . . .	4	25.372	4.09	7739	24.8	do.
Blackwater .	4	25.352	3.00	7237	33.8	do.

TABLE II.

HALF SPEED COAL CONSUMPTION TRIALS.

Name of Ship.	Duration of Trial in Hours.	Speed in Knots.	Water Lost in Tons.	I. H. P.	Total Coal Used, in Tons.	Kind of Engine.
Eden . . .	12	13.56	?	......	15.55	Turbine.
Exe . . .	12	13.3	.67	730	5.67	Reciprocating.
Itchen . . .	12	13.03	2.5	705	5.63	do.
Usk . . .	12	13.0	1.58	727	5.07	do.

TABLE III.

DIMENSIONS AND OTHER DATA FOR THE FOREGOING SHIPS.

Name of Ship.	Dimensions in Feet.			No. of Screws.	Disp. Tons.	Intended I. H. P.	Mean Speed Expected.
	Length.	Beam.	Draft.				
Eden . . .	220	23	8¾	6	527	7000	25
Exe . . .	225	23½	10	2	540	7000	25½
Itchen . . .	225	23½	10	2	550	7000	25½
Usk . . .	225	23½	10	2	550	7500	26
Blackwater .	225	23½	10	2	550	7000	25½

Table III is from Brassey's Naval Annual, 1903, page 331. Since on the trial sheet given us by the Admiralty the trial displacements were marked as from 570 to 610 tons, all the boats must have been considerably overweight.

The following table gives the coal per I.H.P. per hour for those three ships for which we have both the high and low power,—

TABLE IV.

CONSUMPTION OF COAL IN POUNDS PER INDICATED HORSE-POWER PER HOUR.

Name of Ship.	Pounds Coal Per I. H. P. Per Hour.	
	Full Speed Trial.	Half Speed Trial.
Exe	2.11	1.45
Itchen	2.46	1.49
Usk	1.79	1.30
Mean	2.12	1.41

$$\frac{141}{2.12} = 66.5\%.$$

Thus at about half speed the rate of consumption per horse-power is only 66.5 per cent. of that at full power with the reciprocating engine. These low power figures are by no means the best economy that can be obtained with the reciprocating engine.

Table V gives the coal per mile at the speeds on trial. As it might be objected that the "Eden's" half speed trial was nearly half a knot higher than the mean speed for the "Exe," "Itchen," and "Usk," the approximate coal per mile is also given for all the four ships at the "Eden's" speeds—that is 25.287 and 13.56 knots—the correction being made by assuming,—

1st. The resistance to vary as the square of the speed. This is very approximately true at the lower speed where the correction is greatest; and not quite so true at the higher speeds where, however, the correction is a smaller percentage.

2d. The efficiency of the engines to be constant over the small ranges of the corrections. This is practically accurate for the reciprocating engine for which alone corrections are made.

We must not push the analysis of these trials too far, as we do not know the exact trial displacement and trim of the boats; but the "Eden's" designed displacement was the smallest, and this probably also holds for her trial displacement. Yet, Table V shows that her full speed consumption per mile was just under the worst result given in the table for the reciprocating engine; it is fully 11 per cent. more than the mean result and 22 per cent. more than the best, from the reciprocating engine. This can scarcely be called a good record.

This comparison may not be entirely fair to the "Eden," as her maximum speed is *possibly* a good deal greater than the table gives or the contract calls for, and her economy may improve with increase of speed. But we have still the definite statement by all the high officials of the Admiralty, mentioned above, that the economy at full speed is not better than with the reciprocating engine.

TABLE V.

CONSUMPTION OF COAL IN TONS PER MILE.

Name of Ship.	Full Speed Trials. Coal Per Mile in Tons.		Half Speed Trial. Coal Per Mile in Tons.	
	Actual Speed.	25.287 knots.	Actual Speed.	13.56 knots.
Column	ONE	TWO	THREE	FOUR
Eden	.295	.295	.0956	.0956
Exe	.255	.255	.0355	.0369
Itchen	.296	.297	.0360	.0390
Usk	.244	.242	.0325	.0354
Mean for Reciprocating Engines	.265	.265	.0347	.0371

	Actual Speeds. Columns one and three.		Corrected Speeds. Columns two and four.	
	Turbine.	Reciprocating Engine, from Mean Values.	Turbine.	Reciprocating Engine, from Mean Values.
Increased Economy of Half Speed $= \frac{\text{Coal per Mile Full Speed.}}{\text{Coal per Mile Half Speed.}}$	3.09	7.64	3.09	7.14
	Actual Speeds.		Corrected Speeds.	
Ratio of Increased Economy $= \frac{\text{Turbine.}}{\text{Reciprocating Engine.}}$	$\frac{7.64}{3.09} = 2.47$		$\frac{7.14}{3.09} = 2.31$	
Ratio of Radius of Action at Half Speed per Ton Coal $= \frac{\text{Turbine Consumption.}}{\text{Reciprocating Engine Consumption.}}$	$\frac{.0956}{.0347} = 2.76$		$\frac{.0956}{.0371} = 2.58$	

TURBINE PASSENGER STEAMER "LONDONDERRY"

STERN OF THE "LONDONDERRY"

When we reduce to half speed, the line "Ratio of Increased Economy" shows that the consumption per mile improves from two and one-third to two and one-half times as much with the reciprocating engine as with the turbine; and, as the former started from a rather better economy, the radius of action of the ship fitted with reciprocating engines is from two and one-half to two and three-quarters as much per ton as with turbines.

The "Velox" trials were not completed, but Sir John Durston and Admiral Oram stated that the coal consumption in all their turbine ships had been found most forbidding at low powers.

This difficulty must, quite early, have been felt by Mr. Parsons as, to meet it, he proposed a partial return to the reciprocating engine, which he actually carried out in the "Velox." This ship has four shafts, driven, at high speed, by turbines; but for cruising there are small triple expansion engines attached by a clutch coupling to the forward end of each of the outside shafts.

Admiral Oram says that the partly-completed trials of this ship have demonstrated that this device had been found no advantage at low powers. It has not been repeated by Mr. Parsons who, we understand, now advocates fitting extra turbines for cruising, increasing the number of stages for the steam and thus attempting to compensate for the reduced peripheral velocity. The Allgemeine Electricitäts Gesellschaft also make the same proposal.

Mr. Yarrow, in the ship with Rateau turbines, has fitted three shafts, the two outer alone being driven by turbines, and the center shaft by a small triple expansion engine. The whole installation is about 2,000 I.H.P. and the reciprocating engine about 300 I.H.P. This engine was, we believe, introduced partly to save backing turbines and also for cruising.

We could not predict for either of the latter plans much better success than is attained in the "Velox." In Mr. Yarrow's plan, the propeller of the small engine, if it is to have the proper slip at full speed, will have a very great slip at half speed, and consequently will be inefficient; thus neutralizing the good economy of the engine. Indeed, it obviously has not solved the problem, for Mr. Yarrow stated his belief that the inefficiency at low powers *will* be overcome. Besides, Messrs. Sautter Harlé & Co., who are the principal makers of Rateau turbines, and who knew the results from Mr. Yarrow's boat, spoke decidedly of the large consumption at low powers.

With the turbine alone, we have only to reflect that the area through the blades which is sufficient for full power is enormously large for half speed. If the turbine is to maintain its full-power economy—not improve it at a reduced power, as in the case of the reciprocating engine—the quantity of steam is reduced to about one-tenth. No allowable addition of cruising turbine can come near making the pressure gradients as small as would be required to maintain economy.

The only other proposal with which we are acquainted is that of Geheimer Baurath Rathenau, General Director of the Allgemeine Electricitäts Gesellschaft. He stated to us that he believed the best way to apply the turbine on shipboard was to drive it at high speed, generating a single phase electric current; and by this drive motors attached to the propellers. Then at low power he would still drive the turbine fast. His company had got out a scheme for this, but he said the objection was that the weights came out too great. He thought this objection would be overcome—a conclusion with which we entirely disagree. Putting aside the question of efficiency, the interposition of two transformations of energy between the turbine and the propeller, whether it be this particular scheme or any other, is certain to add a large percentage to the total weight. If the efficiency also suffers, the futility of the scheme is emphasized.

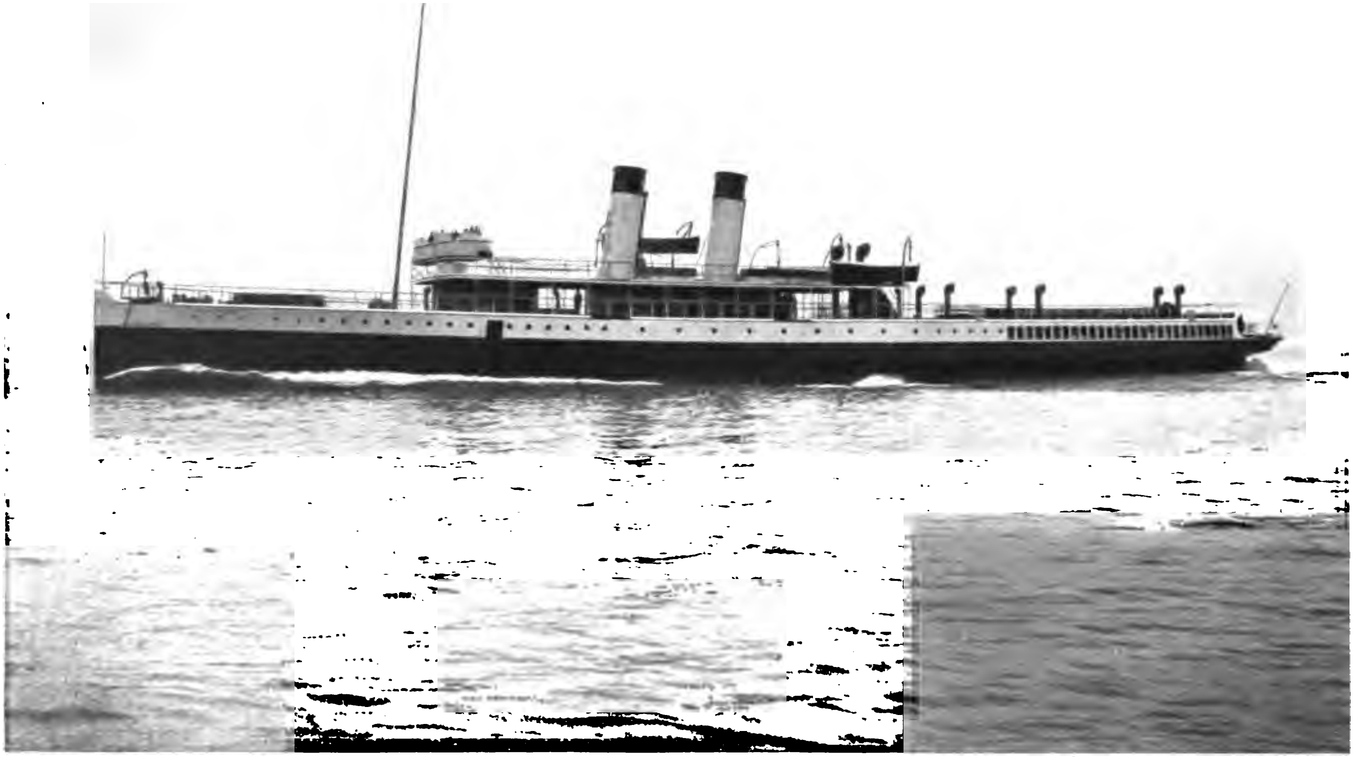
The difficulty of the low-power consumption is an extremely serious one and a practical solution has, in our opinion, yet to be proposed.

Altogether, we do not think the claim for superior steam economy in the marine turbine has been established even at full speed and the only numerical results we have obtained point in the opposite direction. The widely-recognized inefficiency at low speeds is one of the main obstacles restricting its field.

III (b) and (c).—ECONOMY OF ATTENDANCE AND OIL

The turbine has an advantage from the smaller quantities of oil and other stores used. This leads to some reduction in the engine room staff, though we think the reduction made should be small; say, one oiler for a moderately large and two for a very large engine.

The saving of expense in oil in a large ship will, no doubt, be considerable; but the largest advantage will, we think, be in the ease with which the boilers and



TURBINE PASSENGER STEAMER "QUEEN ALEXANDRA" main condensers can be kept free of oil, especially if the steam from the auxiliaries is led into a separate condenser.

On the other hand we think there has been a tendency to make much more of this claim than is strictly just. In the best practice no cylinder oil is used and the only supply from the main engines finding its way into the steam is from the piston and valve rods, which need only be small in amount. Almost all the oil used is put on the outside bearings and has no chance of affecting the feed water.

IV.—BALANCE

The avoidance of severe vibration through the almost perfect balance of the turbine is the claim which has been most nearly fulfilled; and it is a very important one. Dr. Elgar remarked, that if it avoided vibration that was sufficient reason for putting it in the new Cunarders.

The claim was too absolutely made at first. For instance, Mr. Parsons in his 1901 paper, read at Glasgow, which has already been cited, claims "Absence of vibration"; and in a publication by the Parsons Company, the first claim is "Complete absence of vibration from Main Engines," entirely ignoring propeller vibration.

The facts are these. In the engine room of the "Queen" the noise from the turbines was such that it was necessary to shout into the ear of the one you were speaking to, and in parts near the engine room there was a very considerable sensation from this extremely short period vibration. At more distant parts of the ship, however, the sensation was much less and the result exceedingly satisfactory. Admiral Oram's opinion regarding the "Queen," "Brighton," and British naval vessels fitted with turbines, is that the vibration is negligible. However, comparing the "Queen" and the "Cedric" in this respect, we think the latter was much ahead, due, of course, principally to the comparatively small power in the large hull.

The high speed of shaft in the turbine boats has in several cases given rise to very severe propeller vibration, felt principally at the stern. Even with propellers which are perfectly formed and balanced this disturbance would present itself, being caused by the irregularities of speed and direction of the streams through the propeller. In those ships which have more than one propeller on a shaft, the forward propellers are necessarily near the hull and the irregularity of stream action is intensified. It is in these cases that propeller vibration seems to have been most severe. Thus in the "King Edward," when she first came out, the vibration at the stern, and specially in the dining saloon, was very serious; the water appeared to boil in the caraffes, and all the dishes on the table rattled. Indeed, instead of being freest from vibration of any boat on the Clyde, her vibration was probably the most excessive. Forward of amidships the vibration was practically insensible, and in the "Queen Alexandra," brought out next year, the first-class quarters were, consequently, placed forward. We have heard that now the "King Edward" has only one propeller on each shaft and, if so, we would expect that the vibration would be much diminished. Whether this information is correct or not, the "Queen Alexandra" had only three propellers fitted.

Again, we are informed on good authority that the "Tarantula" vibrated so excessively that the rivets had repeatedly slackened in the stern plates and frames near the propellers.

But the simple cure for much the larger part of this propeller vibration is to have only one propeller on each shaft; and this, happily, has been found to be the best arrangement for efficiency.

The claim that has been put forward for extra safety from absence of reciprocating parts we think of little importance as, even in torpedo boats, not many accidents have occurred and in other ships the risk is extremely minute. The bursting of a turbine is not an impossibility and would be a risk of about the same order as the former.

V.—REDUCTION OF PROPELLER DIAMETER

In determining the size of a propeller sufficient surface must be provided to prevent such great pressure per square foot of surface as will produce cavitation at the highest speed, due to excessive pressure gradients. The same total surface, practically, must be there whether only one screw is used or a number. Thus, decrease of propeller diameter comes from the increase of the number of screws. It is not due directly to the turbine. If three shafts with one propeller on each are used, as we think best for large passenger steamers fitted with reciprocating engines,—and as most turbine steamers are now fitted,—the same advantages will be attained whatever the motive power is. But with the turbine we would frequently have three, or even four, shafts, where with reciprocating engines almost every naval architect would fit two shafts, or even only one, for simplicity and lower first cost. Thus, while it was to gain the advantage of reduced propeller diameter that naval architects were forced to adopt twin screws in large passenger steamers, on account of the shallow water over the bars of important harbors, and while that was an advantage seen and claimed for the triple screw warships of the American and German navies, it has naturally been brought into prominence again by the triple or quadruple shafts of turbine steamers.

We had an excellent opportunity of witnessing the effects of reduced propellers, as on 9th February of this year we made the double crossing between Dover and Calais, on the "Queen,' in a full gale. The turbines did not race, while with an ordinary size and immersion of propeller we have no doubt the racing would have been very severe.

VI.—REDUCTION OF INITIAL COST

We are not aware that so far there has been any reduction of first cost, but we have no doubt that when a healthy competition arises, as it is sure to do soon, this advantage will be added to those of the turbine.

VII.—REDUCTION OF COST OF UPKEEP AND OVERHAUL

We feel it is too soon to reach any definite conclusions regarding the probable reduction of cost of upkeep and overhaul. The great reduction in the number of bearings from those required for a reciprocating engine should greatly reduce the cost of adjustment. On the other hand, one necessity of the turbine is that the blades should run very near fixed parts of the machine, and there is some liability to the stripping of blades. This has occurred on land and, we believe, also in marine turbines. Such an accident might lead to a costly repair. As the effects of expansion from heat will be more manifest the larger the turbine, this danger becomes greater the larger the installation. Whether it will prove important or not, experience alone can prove.

BACKING AND MANEUVERING

It can, unfortunately, be urged against the turbine that it cannot be reversed. Though many attempts have been made, no practicable form of reversing turbine

TURBINE PASSENGER STEAMER "QUEEN"

BOW VIEW OF THE "QUEEN"

has yet been devised. Hence, on shipboard, one or more backing turbines must be used, which are usually in the same casing as the ahead turbine; or, as in Mr. Yarrow's boat with Rateau turbines, a reciprocating engine may be used on a separate shaft for going astern.

The consumption of steam in backing with turbines is also very great as, to save weight and cost, the backing turbines are made small and inefficient. For instance, the Chief Engineer of the "Queen" told us, when backing slowly out of Calais harbor, that he was using more steam than when going full ahead. Indeed, what caused the inquiry was that he increased the forced draught pressure then above that for full ahead. This inefficiency in itself, however, is not a serious objection; for, as the actual time the engines are backing is very small compared with the time they are going ahead, the running cost will not be much increased.

The comparatively poor maneuvering power of turbine-driven steamers is generally acknowledged and arises from at least two causes. If the outer shafts only are used when maneuvering, as will probably be general either with three or four shafts, the small propeller surface in play will give correspondingly diminished control; that is, it will affect the starting, stopping, backing, and turning with the propellers. Again, the very deficient economy in backing, just noticed, will make it difficult to stop and back, or perform evolutions quickly, if the ahead speed for some time previous has been slow; for then the fires will not be in a condition to give nearly the full supply of steam.

This case should occur in dense fogs, though "half speed" then seems to be a fiction of the log book; but so much the worse for the ship which has to depend on part of its propeller area for stopping. It certainly will occur constantly in warships maneuvering together, whether in peace or war. Indeed, this faulty maneuvering power led Herr Kaiserlich Marine-Baumeister Gustav Berling, of the German Navy, to remark to us that he hoped all the ships of the enemies of Germany would have turbines.

Mr. Yarrow's arrangement for backing with the center screw alone we consider very bad, as not only is there only one small propeller in use, but the advantage of twin screws is almost entirely thrown away.

The deficiency of the "King Edward" in maneuvering was evident; as, when she came out at first, especially, she had great difficulty in making the piers. Also, the manager of the Continental Department of the London, Brighton, & South Coast Railway Company told us that the backing and maneuvering qualities of their steamer "Brighton" were poor.

ARRANGEMENT OF TURBINES AND PROPELLERS

With regard to the number of shafts which it is best to use, we received opinions from a good many, and usually the preference was for three. Most of the turbine ships that have been built, and all those building at Messrs. Denny's, have three shafts; while the German vessel just launched at the Vulcan Works in Stettin and also, we believe, the large Cunarders, will have four. The Allan Liners are, we believe, designed with three shafts, while the yacht "Revolution" has two.

Mr. Philip Watts, Sir John Durston, and Admiral Oram, expressed their preference for three shafts; while Prof. Dr. Phil. Klingenberg, Director of the Allgemeine Electricitäts Gesellschaft, indicated that his company advocated two shafts with more than one propeller on each, but so far they have done no marine work. Mr. Yarrow, as noted above, believes it is best to drive two outer shafts by turbines exerting nearly the whole power at full speed, and actuate the center shaft by a small reciprocating engine. Professor Rateau advocates the same system,

TURBINES OF THE "QUEEN ALEXANDRA"

THE FIVE PROPELLERS OF TURBINE YACHT "EMERALD"

except that instead of leading the exhaust of the reciprocating engine to the condenser, as Mr. Yarrow does, he would lead it to the L. P. turbine, or, in some cases, even to the H. P. turbine. This treatment of the exhaust would certainly not do when backing. He thinks the reciprocating engine may even be designed for half the whole power. This, however, can hardly be admitted among turbine systems.

The question depends primarily on the propeller and the requirement of comparatively high peripheral speed in the turbine to attain economy. The propeller cannot be made economical if the pitch-ratio is less than 1, or, at the outside, .8. The more propellers there are, the total projected area being constant, the smaller will be the diameter and the faster the revolutions for a given pitch-ratio. Therefore, the greater the number of propellers the smaller may be the diameter of the turbines for a given peripheral speed. If one propeller were used the turbine would be large, heavy, and difficult to stow, and the ship would lose the advantages in safety and maneuvering arising from twin screws and the previously discussed advantage of smaller propellers. There would, of course, be the counterbalancing advantage of simplicity of construction and the whole propeller area would come into play in stopping and backing.

If two shafts are adopted the turbines will be much smaller in diameter and a good deal will be saved in weight.* The turbine centers will be lower and consequently the shafts can be laid off more nearly horizontal. The screws are much better immersed for a given draft with the corresponding decrease of lia-

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* Comparing the cases of one and two shafts with one propeller on each shaft, the propellers being similar in design, with the same pitch-ratio and slip we have

Propeller diameters as $\sqrt{2} : 1$

$\therefore$ Revolutions as $1 : \sqrt{2}$

$\therefore$ Diameters of turbines, also of similar design and the same peripheral velocity, as

$$\sqrt{2} : 1$$

$\therefore$ Weights of one turbine and propeller in each case as

$$2^{\frac{3}{2}} \quad 1$$

$\therefore$ Total weights of turbines and propellers in each case as $2^{2} : 2 = 2^{\frac{1}{2}} : 1 = 1.41 : 1$.

Similarly, comparing a two-shaft and three-shaft system, we have the totalweights as

$$3^{\frac{1}{2}} : 2^{\frac{1}{2}} = 1.2 : 1,$$

And for three shafts against four

$$4^{\frac{1}{2}} : 3^{\frac{1}{2}} = 1.15 : 1.$$

No doubt the added complication, as the number of shafts increases, would sensibly diminish the advantage in weight.

bility to race; the ship has good maneuvering power and still the full propeller area for stopping and backing, as the turbines would be independent and a backing turbine fitted on each side. The added complication is not great.

It seems to us that this arrangement of two shafts might be practicable, especially where the ships are very fast. It was not adopted by Mr. Parsons on the "Turbinia" or any of the other ships he has engined, probably because he found that to make this type of turbine economical, with comparatively low speeds of revolution and allowable diameter, it had to be made exceedingly long. But if any of the shorter varieties of turbine come to the front for marine work, we expect that twin screws with quite independent motors will frequently be preferred. This has already been partially indicated by the fitting of only two Rateau turbines in the boat designed by Mr. Yarrow. The third shaft, driven by a reciprocating engine, would in all probability never have been added but for the necessity to attempt to reduce the consumption at cruising speed. And in a still earlier installation of Rateau turbines in a French torpedo boat, Messrs. Sautter Harlé & Co., the makers, told us they had adopted only two shafts.

The adoption of three shafts driven by turbines increases some of the foregoing advantages: the propellers and turbines are of smaller diameter, somewhat reducing the total weight; stowage so far as height is concerned is easier, though horizontally it is more crowded, and the center turbine will frequently have to be placed ahead or abaft of the side ones; it should be possible to make the design on rather less weight, though there will not be much margin; the immersion of the propellers for a given draft is greater. On the other hand, if the turbines are fitted as Mr. Parsons has always done—dependent, with astern turbines only on the outer shafts—only two-thirds of the propeller area is available for stopping, backing, and maneuvering, which, experience has shown, leads to a very marked inferiority in these respects. We also lose, what is most essential for warships and a considerable advantage in other cases, namely, the power of going ahead with one screw and astern with the other; and thus, if necessary, turning the ship on its own center. Also, the complexity is decidedly increased.

One disadvantage noted by Mr. Leslie Denny as peculiar to triple screws,—indeed, stated by him as the principal disadvantage—is the wash of the center screw against the rudder post. This, of course, is much more serious with three than with one screw, as a much larger proportion of the smaller propeller is masked. This disadvantage was reduced, in some of their ships, by stopping the rudder post short and overhanging the rudder below it.

This arrangement of triple screws is adopted as a way of escape out of a great difficulty and is not to be confused with the triple-screw system so strongly to be recommended for warships and large passenger steamers. In triple screw warships fitted with reciprocating engines the units are quite independent, greatly diminishing risk of total disablement, and improving the economy at low powers even over twin screws; the advantages to be derived from small diameter of propeller is preserved and all the propellers are available for stopping and backing, with full efficiency, while two-thirds of full power can be used to assist steering. With a short variety of turbine, independence of the units, in the case of three screws, would without doubt be arranged, with greatly increased advantage over the present system of a single unit, in which the risk of total disablement is very large.

A four-shaft arrangement goes further in the same direction as three shafts, though no sensible saving of weight need be looked for; we would rather expect a slightly increased weight arising from the increased complication. Its only distinct advantage over three shafts with dependent units, is that we now have two quite independent units with the well-known advantages resulting therefrom. The complication is so great that to decide between this and three shafts seems rather to be a choice of evils than a choice of advantages.

With short turbines, that could be run efficiently at a moderate peripheral speed, we would most strongly advocate two shafts for small ships and three shafts for large, the units being quite independent.

The allowable speed of revolution of the shafts naturally must decrease as the power of the installation and the diameters of the propellers increase. To gain all possible advantage of economy and weight from the turbine, it was natural to begin with very high speeds of revolution in the "Turbinia," but these have had to be materially reduced. Thus, Admiral Oram told us that in the earlier boats for the British Admiralty—presumably referring to the ill-fated "Viper" and "Cobra"—the speeds of revolution had been 800 and 900 per minute, but their practice now is about 500 per minute—presumably in the "Velox" and "Eden." Messrs. Sautter Harlé & Company, in the torpedo boat, mentioned above, for the French Government, with Rateau turbines of about 2,000 horsepower, adopted a speed of revolution of shaft of 1,700 to 1,800 per minute. They seemed to look on this boat as a complete failure and ascribed the high speed of revolution as the cause, giving a very poor performance of the propeller. The superintendent said they know now the revolutions must be much slower, and in Mr. Yarrow's boat they designed the turbines and propellers for much smaller revolutions.

We were informed by the chief engineer of the "Queen"* that the center screw had 480 and the side screws 500 revolutions per minute, the center propeller being six feet diameter and the side ones five feet eight inches, each with three blades and of manganese bronze.

Before the "Turbinia" became a success, Mr. Parsons had adopted more than one propeller on at least the side shafts, and this for some time was the common practice and was believed to give a better result by avoiding cavitation. We believe it was during the experiments on the yacht "Emerald," on the Clyde, that doubt was first thrown on this; and, after removing the forward propeller on each shaft, it was found that the yacht performed better and was much freer from vibration. Subsequent experience has fully confirmed this, as stated by Mr. Parsons in the discussion of his paper before the naval architects in London last year.

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* We believe that these data respecting the "Queen" differ from some which have been published. The above figures were merely given in conversation.

Admiral Oram's statement to us was to the effect that more than one propeller on a shaft was found no advantage—that they were at the experimental stage with propellers and had yet come to no other conclusion except that there should only be one per shaft.

Mr. Leslie Denny said that all the seven ships they were building had only one propeller on each shaft. This he thought usually the best arrangement, though there might be cases where two would be an advantage.

The owners of the "Queen" informed us that she had originally five propellers, but that they had removed the forward ones on the side shafts, leaving her now only with three. We also understand that the "Queen Alexandra" and "Brighton" have only three.

Prof. Flamm, of Charlottenburg, has made many tank experiments with propellers and he told us he had found the best results with one propeller on each shaft.

Mr. Ridsdale, Naval Architect with Messrs. Brown, Boveri & Co., at Baden, Switzerland, was of the same opinion, though he believed that in special circumstances (which he did not specify) two might be better.

TURBINE PASSENGER STEAMER "LOONGANA"

TURBINE PASSENGER STEAMER "PRINCESS MAUD"

The only opinion on the other side which we got was that of Prof. Klingenberg, Director of the Allgemeine Electricitäts Gesellschaft. As stated above, this company seemed to favor two shafts with multiple propellers on each, but they have no marine experience.

The only conclusion to be drawn from the foregoing is that those who have had access to the details of the results of trials have been very strongly and uniformly impressed by the superiority of one propeller per shaft; and in no case, so far as we heard, after trying the single propeller, has there been a reversion to more than one. It may be noted that in making the change there was no increase of the after propeller—the forward propeller was merely removed and the same after propeller put back in place. The change could harldy fail to make a marked diminution in vibration, and we understand this was found to be the case. On what the belief of Messrs. Denny and Ridsdale—that in some cases two propellers per shaft might be better—is founded, we do not know, and have grave doubts as to its correctness. The action of the forward propeller puts the column of water in rotation with sufficient angular velocity to seriously interfere with the action of the after propeller, which is rotating in the same direction. To increase the action of the after propeller on the water we would, consequently, require an increased pitch, with, of course, a correspondingly reduced component of force in the direction of the ship's motion, which means a reduced efficiency. We believe it is due to this action that no advantage has been found from the double propeller.

OPINIONS AS TO THE FUTURE OF THE TURBINE

We, of course, got many opinions from well-known men as to the future of the turbine, in its application to marine propulsion, which it will be interesting to set down.

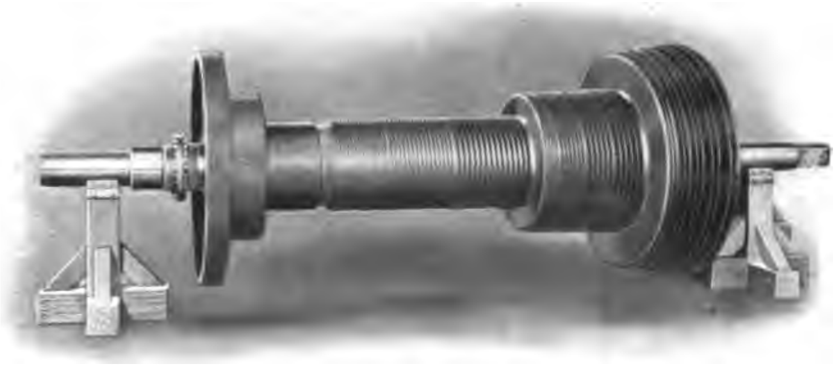
Mr. W. H. Maw, of "Engineering," thought that the turbine had possibly a future both for fast boats and those of moderate speed. But he seemed to think the reversing and propellers might be difficulties.

Mr. Philip Watts strongly believed in its future, and has identified himself very closely with it since long before he left Elswick.

Mr. MacFarlane Gray reserved his judgment. He believed it had a possible field in high speed ships. The rest he had not considered sufficiently, but feared the leakage telling on low powers.

The Hon. W. J. Pirrie expressed his disbelief in the turbine very strongly, though his recent inquiry of the British Westinghouse Electric and Manufacturing Co. seems to indicate a possible change of opinion. He said—thinking, we believe, principally of the Parsons turbine—that its construction did not commend itself to his mechanical judgment. He had advised the British Admiralty to put it into very fast ships—torpedo boats—for trial, as the only possible field. He spoke very strongly of the possible blow-through of steam, and thought that Mr. Parsons was being entirely misled in applying it to low speeds—meaning, we think, the Allan Line Boats.

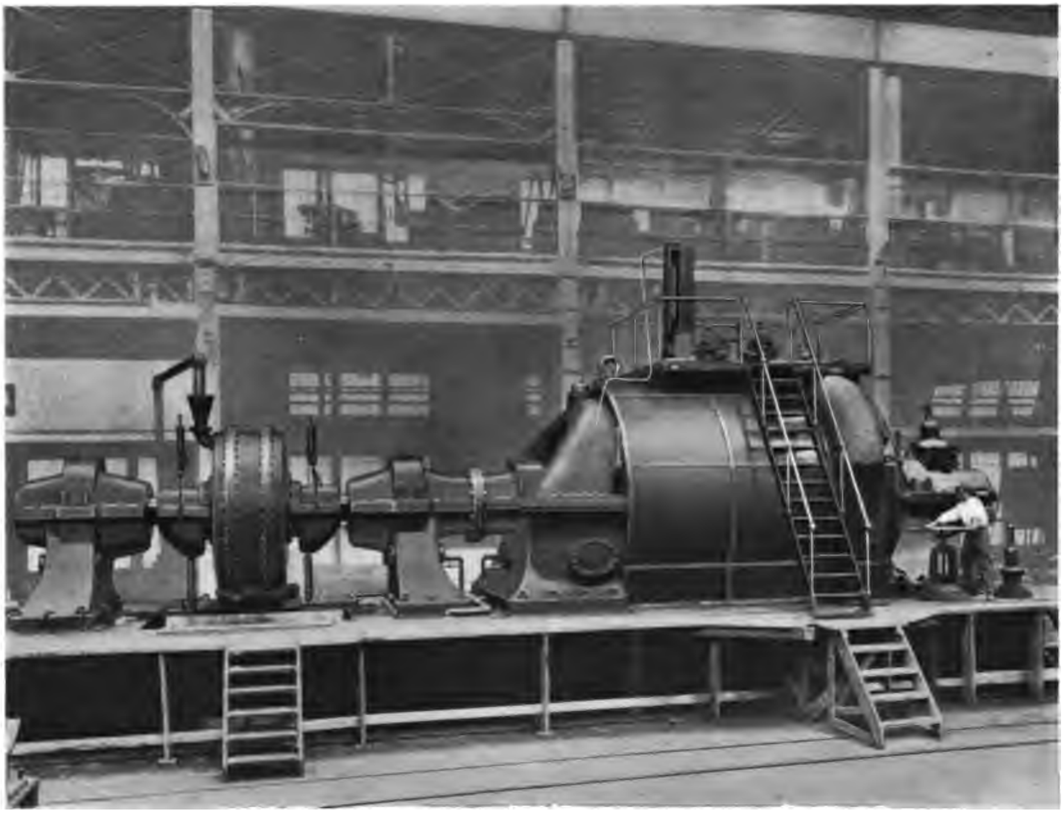
Admiral May believed in the future of the turbine, but said that the most essential test was the coal consumption at low powers when cruising. So far he



REVOLVING ELEMENT—WESTINGHOUSE-PARSONS TURBINE believed that to be fully double the consumption with reciprocating engines. He thought the difficulty would be overcome.

Mr. Yarrow believes that the large consumption at low powers will be overcome.

Admiral Oram expressed the opinion that the turbine would *never* be suitable for warships on account of the inefficiency at low powers, and also would never be suitable for tramp steamers.



7,500 H. P. WESTINGHOUSE-PARSONS TURBINE UNDER BRAKE TEST

Sir John Durston entirely agreed with Admiral Oram on this point, as well as on all others regarding the turbine.

Dr. Elgar was exceedingly dubious regarding the future of the turbine.

While the Report of the Cunard Commission was in favor of turbines, we gathered a very distinct impression from those we met who were likely to know, that is was far from a strong recommendation. As Sir John Durston—who was in close touch with many members of the Commission and no doubt was more or less acquainted with their views—expressed it some three days after the signing of the Report—the Commission saw no objection to trying the turbines in the new Cunarders, but that they did *not* recommend it. Dr. Elgar distinctly stated that he thought they could have done as well with reciprocating engines; but, he added, if it did nothing but avoid vibration it would be worth while adopting it.

At Hamburg we found the most emphatic opposition. Messrs. Blohm and Voss, Mr. Frahm, and Mr. Berling, all united in condemning it on account of the propeller difficulties, the bad efficiency at low powers, and the bad maneuvering qualities. As noted above, Mr. Berling hoped that all the ships of all the enemies of Germany would have turbines. They were evidently well acquainted with the low-power results obtained by the British Admiralty trials.

Mr. James Sulzer did not believe in the turbine for steamships and was also impressed with the difficulties regarding propellers.

We were interested to hear an opinion from a member of the Denny firm, as they have enthusiastically advocated the turbine for several years.

Mr. Leslie Denny thought it could only be applied to ships which had to do all their work at high speed. Its application to tramps, or ships which had to run much at low speed, was not yet in sight. None of the seven turbine ships they were building were under 17 knots speed and he thought this is about the lower limit for advantage with the turbine—15 knots was folly.

Altogether, the impression made on us was one of great reserve on the part of those most intimately connected with the turbine, amounting in some cases to positive distrust. While many had an open mind as to the possible great future of the marine turbine for high speed ships, there was no echo of the popular and uninformed judgment, that a great revolution in marine propulsion is certainly immanent. There is only one statement that we have seen where the turbine has been put forward as a possibility for slow speed ships, and this is by the Hon. Chas. Parsons in the discussion of his last year's paper read before the Naval Architects, London, where he says,—

"I may say we should be quite ready to deal with a set of engines of 10,000 h. p. and 10 knots speed, which is just on the line of demarcation, ... "*

In view of the known facts, this statement seems to us perfectly astounding, coming from a man who has been the leader in what has been already done. And we cannot believe that it is meant literally, as we know that he resented an inquiry, by a well-known firm of ship builders, to put the turbine in a ten-knot ship.

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* It is perhaps better to give the quotation more fully. Though we think it all open to criticism, the latter part is not of the extreme character of that given in the text,—

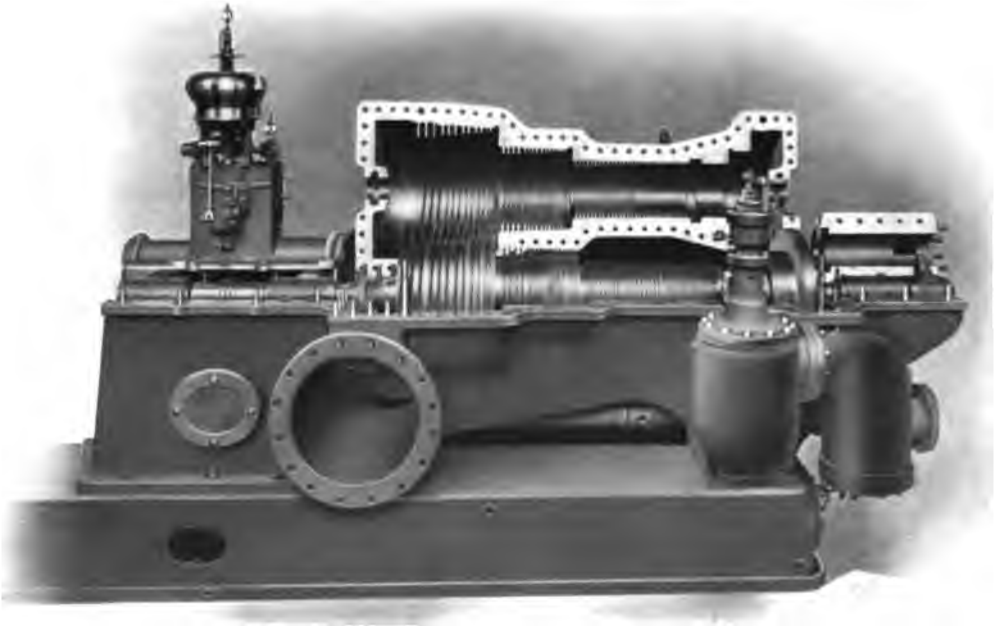
" In regard to the present limits to the application of the steam turbine, I may say that we should be quite ready to deal with a set of engines of 10,000 h. p. and 10 knots speed, which is just on the line of demarcation, but it is very much easier to design efficient turbine engines for a vessel of 20 knots and 10,000 h. p., and the advantages would be very much greater in the latter case. The line of demarcation would run at present at about 18 knots and 2,000 h. p., and 15 knots and 5,000 h. p., and 12 knots and 10,000 h. p. There are special cases of yachts and shallow-draft vessels, where considerable advantage over ordinary engines would result in the case of even smaller horse powers, and slower speeds than these."

We are rather inclined to agree with the position taken up by Mr. Alfred Morcom in the discussion of the same paper; who, while believing the turbine is bound to go on, puts in a strong defence of the admirable qualities of the reciprocating engine, which have been unjustly reflected on by supporters of the turbine.

If the inefficiency at low powers and speeds of revolution cannot be overcome, Admiral Oram's prediction that the turbine will never be suitable for warships will certainly be fulfilled.

In warships the largest current expense, exclusive perhaps of the pay of officers and crew, is the coal bill, and nine-tenths of the coal is burned at cruising speed. The increase of the coal bill by a good deal over 100 per cent. would alone be a sure preventive of the introduction of the turbine in war vessels. But that is the least serious aspect of the question.

One of the most valuable qualities a warship can possess is large radius of action. If she cannot, on occasion, go a long way to meet the enemy, or keep the



400 K. W. WESTINGHOUSE-PARSONS TURBINE OPENED FOR INSPECTION

sea a long time while watching for him, her otherwise admirable fighting qualities will often be rendered quite useless. Any system of propulsion which would reduce this radius of action one and one-half times, not to speak of over two and one-half, as Table V shows, cannot possibly be thought of.

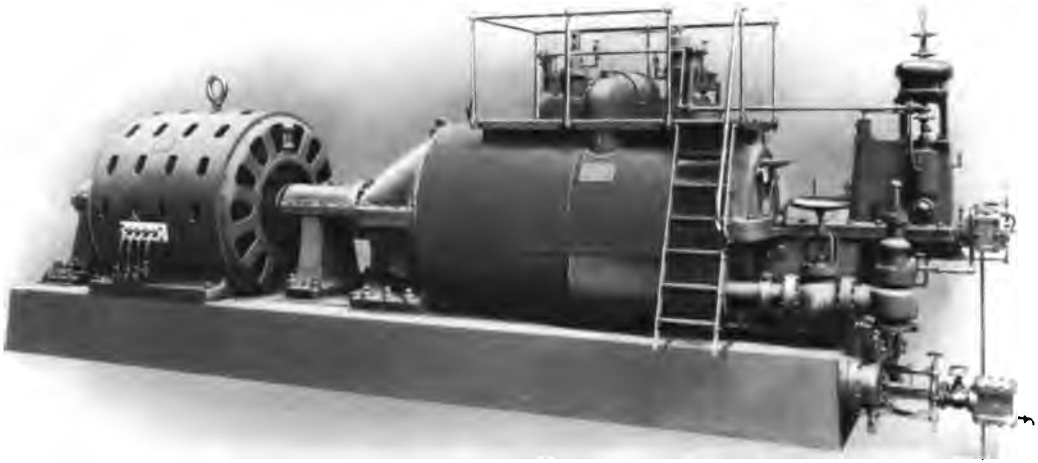
This seems at once to limit the field to fast ships, which, when running, have to go all the time at full speed. Under the best conditions, what has been said precludes us from believing that very considerable saving can at present be looked for even then. Therefore, a fast ship which had to run any considerable fraction of her time at reduced speed,—for instance, if she had a short open sea passage, and had to run slow in a long estuary or river—would have her high speed economy quickly wiped out and reversed.

This slow speed is a condition which not infrequently happens with all ships. For instance, in thick fogs a fast ship should greatly reduce her speed and in heavy head seas she must do so. Again if, through delays or other causes, coal is running short, a great saving is effected and port easily reached by easing the engines; but with turbines, Table V shows that this would be very much less effective.

We are inclined to believe with Professor Rateau that the lower limit of speed for the effective use of the turbine is not under 20 knots, and we think it may possibly be higher. In fact, we do *not* regard the marine turbine as yet past the experimental stage. But if it did prove appreciably more economical even at 20 knots, there is a wide future for it.

No doubt, if the efficiency at lower powers could be much improved, a greatly broadened field would at once open. But those who stated their belief that this improvement would be made did not even attempt to give any ground on which their belief was based. It seems to us that the teaching of theory is perfectly plain and that the economical turbine is one of large peripheral speed unless the number of stages is increased beyond all allowable limits; and even then other causes of loss would enter to prevent the desired result being realized. Hence in slow ships the diameter and weights must be made large or the propeller must be made to run fast. The first alternative is barred by the dimensions and displacement of the ship and the second by the greatly reduced efficiency of the propeller.

Again, the turbine is essentially a leaky machine, and this alone would greatly reduce the economy when the turbine is run slow, if there were no other cause; but the now reduced peripheral speed and the very excessive area for the diminished quantity of steam, strongly contribute to the same result. We cannot see any reason to hope that a door will open in this direction, to effect economy in fast ships when running at cruising speed. Nor, as already explained, do we believe any of the combinations proposed by Messrs. Parsons, Yarrow, Rateau



750 K. W. WESTINGHOUSE-PARSONS TURBINE UNIT, COMPLETELY ASSEMBLED

or Rathenau, will be found effective or permissible. In what other direction a solution is to be looked for we are at a loss to imagine.

The forms of turbine which have been invented in the past and that are now coming out each year, are very numerous, and it requires no insight to know that the recent applications of the turbine and the large attention which has been drawn to the subject will produce an abundant crop in the immediate future. There is one condition that immediately affects the thermodynamic efficiency of any design of turbine which we will discuss very shortly; as, if this condition is not reasonably approximated to, the particular design cannot meet with commercial success, however good it may be mechanically.

The turbine is a heat engine and the limit of thermodynamic efficiency when working between the absolute temperature T of the inlet steam and T_1 of the condenser, is given by the well-known expression

$$\frac{T - T_1}{T}.$$

This means that each foot-pound of energy is extracted from the steam at the highest temperature possible. Or, in other words, that the expansion from T to T_1 is adiabatic and that the energy of translation of the steam is at all times negligible.

This limit is not reached by any turbine and for three main causes,—

1st. Radiation; which we may neglect, as, by proper lagging, it can be reduced to a very small amount. We might also include conduction, transferring heat from hotter to cooler steam, but probably this leads to a loss which is small in amount.

2d. Leakage of the steam past the surfaces on which it should act. This is well recognized and has been sufficiently referred to above. It probably affects the "action" turbines, such as the de Laval, Rateau, and Zoelly, less than the "reaction" type, such as the Parsons.

3d. Vortex motion or eddying of the steam, which we particularly wish to refer to here.

In a cyclone we have intense vortex motion and the transference, often for long distances, of a very large quantity of energy by the rotating air, with little effect on that immediately surrounding, the energy only very gradually being absorbed. The effect on obstructions is great only when they lie very near indeed to the path of the vortex center. If we could suppose the same amount of kinetic energy possessed by a mass of air devoid of rotary motion,—that is, with a stream line motion—the result would be very different. The still air which lay in front of the storm, or any solid obstruction in the belt swept by it, would be powerfully acted on, with the result that the energy of the originally moving air would be almost suddenly absorbed.

Thus it is, to some extent, in all turbines; the motion is by no means purely along stream lines, but the more nearly this character of motion is attained the greater will be the thermodynamic efficiency of the machine.

If, through roughness of the surface of the steam passages, or too sudden enlargements or contractions making corners and pockets, vortex motion is set up strongly, the thermodynamic efficiency will be lowered as energy can then be readily transferred past the portion of the cycle at which, if maximum efficiency is to be realized, it should be transformed into mechanical effect on the moving blades.

Turbines, such as the Rateau and Zoelly, where the steam is delivered successively through a series of chambers, no doubt have their efficiency partially preserved by the check this vortex motion receives at each stage. During the very brief time the steam is in one of the chambers, expansion having ceased and the pressure being practically constant, the energy of the vortex must be dissipating itself, and the steam will once more have an approximate stream-line motion when expansion recommences in the next series of nozzles.

With turbines of the Parsons type, where the delivery of the steam is always through passages which are virtually nozzles, we must rely solely on the good shape of the passages, and the high finish of the surfaces.

Eddying will also be set up by the rapid relative motion of two surfaces with a thin layer of fluid between; or, to a much smaller extent, by the rotating part moving in the working medium even if well removed from any of the fixed parts. This produces a brake action, more intense the denser the medium, in which energy is actually handed back to the steam. This also will be greatly augmented by want of polish of the moving surfaces.

The viscous resistance thus produced is not to be confounded with that for which co-efficients of viscosity of steam and gases are given in physical text-books. This latter is due to a pure shearing action and the intensity of the resulting force is proportional to the first power of the rate of shear. But in the action we are discussing, the intensity of the force is proportional to the second, or sometimes a rather higher, power of the speed of relative motion of the opposing surfaces and at high speeds rises to be a very important factor. Thus, for this reason alone, we would have little hesitation in predicting failure for both the Wilson and Buchholz turbines, described in the sequel; and we were told that in very high peripheral velocities adopted by Mr. Zoelly at first, the loss from this eddying resistance was quite serious and led him to reduce the speed.

The extremely high finish of all steam passages in the Riedler-Stumpf turbine was a result of the experience of the Allgemeine Company that otherwise great loss resulted.

We believe that avoidance of vortex motion by high finish, smoothness of contour of passages and absence of large surfaces close together with rapid relative motion, are the conditions that should be first looked to in examining a novel design of turbine.

LOWER LIMIT FOR SPEED OF TURBINE SHIPS

In the foregoing, two opinions are given as to the lower limit of speed of ship at which it is advantageous to apply the turbine, and Professor Rateau in his recent paper in London also expresses a very definite opinion.

Mr. Leslie Denny thinks 17 knots is the lower limit at which application of the turbine is advisable, and that 15 knots is folly. Professor Rateau says,—

"There is, therefore, a lower limit of speed, below which the use of turbines cannot be recommended. The author has already expressed the opinion (in a paper read before the Association Technique Maritime in 1902) that this limit is in the neighborhood of 20 knots. The author is aware that certain ships now under construction for transatlantic service, and of a proposed speed of 17 knots, are being fitted with turbine engines, but the future will show how these will turn out."

Mr. Parsons on the other hand thinks that as the power of the installation is increased the lower limit of speed of ship in which it is advantageous to apply the turbine, quickly falls; so that while 2,000 horse-power turbines may be put in an 18 knot ship, 10,000 horse-power turbines may judiciously be put in a 12 knot ship.*

We do not exactly agree with any of these opinions, but we think the two first, giving a constant speed limit independent of the size of ship, are much nearer the truth than Mr. Parsons' belief.

The question is one of great importance in the application of the turbine.

We will discuss it briefly on two approximate assumptions,—

1st. That the maximum allowable pressure per square inch on a propeller is constant; that is, the pressure above which cavitation will occur.

2d. That the Admiralty co-efficient,

$$\frac{V^3 D^{\frac{2}{3}}}{IHP} = \text{Const.},$$

is true.

Suppose two turbine ships which are similar in all respects, both with regard to hull and machinery, and that the linear dimensions are as

$$1 : n.$$

Also let the speeds be the same for each ship. We will have

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* See the quotation in the footnote, page 43.

Resistances as $1 : n^2$.
Propeller surfaces as $1 : n^2$.

Hence, there will be the same pressure per square inch on the propellers of both ships.

As the speeds of the ships are the same and also the pitch ratios of the propellers, while the propeller diameters are as

$$1 : n,$$

the revolutions will be as

$$n : 1;$$

and the turbine diameters being as

$$1 : n$$

the peripheral speeds will be the same.

Thus with the same boiler pressure, the powers developed will be as

$$1 : n^2,$$

or in proportion to $D^{\frac{2}{3}}$ as required by the Admiralty Co-efficient. Hence both turbines and propellers are correctly proportioned and, since the peripheral velocity is the same for both turbines, the efficiency will be practically the same unless the ratio 1 to n is very large. That is, if the application of turbines is wise in one of the ships it is so in the other.

This supports the opinions of Prof. Rateau and Mr. Leslie Denny. But it is modified somewhat by the inaccuracy of the Admiralty Rule. For the larger vessel will be on a relatively lower part of its speed curve, and the wave resistance will be a smaller proportion of the whole. Also, due to the greater length of the ship the skin friction will be slightly reduced, as shown by Froude's experiments on planks of different lengths. For both reasons the second assumption is somewhat in error and the total resistance of the larger ship is rather smaller than the above estimate. Therefore, the propellers could be smaller and, with the same peripheral speed, the ship might be a *little* slower. But for the powers and speeds specified in the quotation from Mr. Parsons, the wave-making resistance is not a serious factor and the Admiralty Co-efficient is not greatly in error. Consequently his statement as to the great drop in limiting speed as the power goes up is very far beyond the truth.

If the allowable pressure on the propeller rises somewhat as the speed of the ship is increased, both vessels would have equal benefit, and a slight inaccuracy in the first assumption scarcely affects what has been said. The larger vessel will, however, get a slight advantage from the greater immersion of its propeller and a consequent allowable increase of pressure per square inch. To offset this, the smaller ship may, with equal safety, be made of rather lighter scantlings than the larger.

The advantage increases as the speed of propulsion rises. For, suppose, by decreasing the dead-weight carrying capacity and increasing the machinery, one of these vessels has her speed increased in the proportion

$$1 : v.$$

The Admiralty Co-efficient being correct, the resistance will increase as

$$1 : v^2,$$

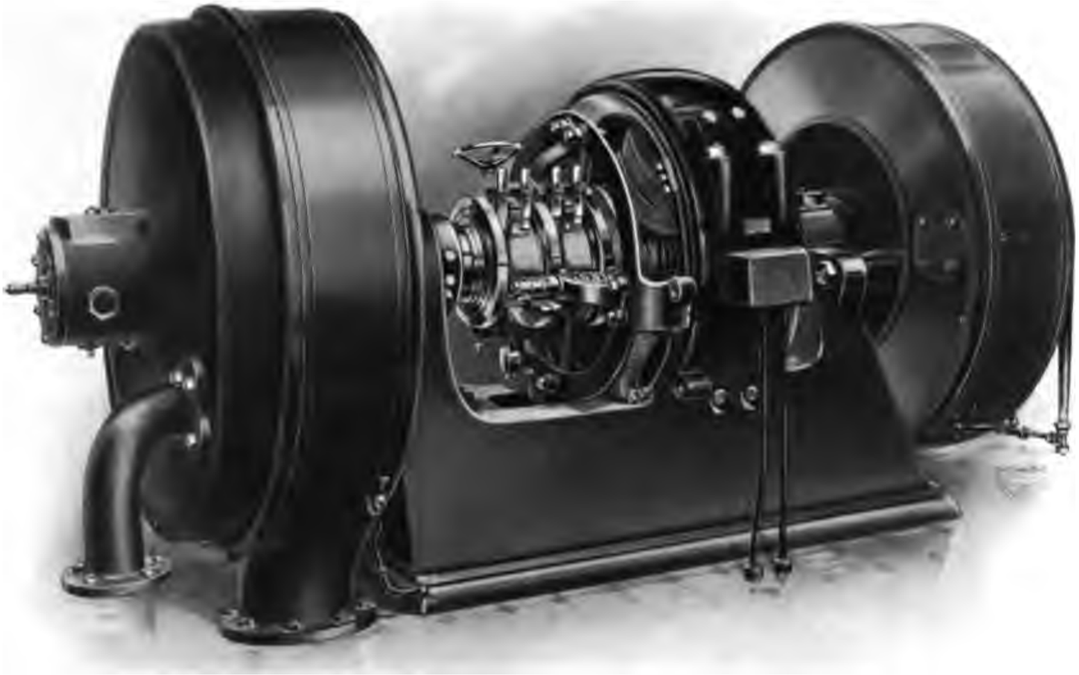
and the projected area of the propeller must rise in the same proportion. That is, the diameter increases as

$$1 : v.$$

If the pitch-ratio and slip are the same as before, the revolutions will also be unchanged. Hence the larger turbines required, if the increase of speed is in any way considerable, will have a larger peripheral velocity, which will give greater economy. But if the increase of speed is in any case pushed to a part of the curve where the Admiralty Co-efficient is very incorrect and the resistance rises much more quickly than the second power of the speed, the rate of increase of advantage will be much diminished. Hence, we reach an important conclusion:—If we are right in believing that the lower limit is not under twenty knots, the ships which the turbine can be applied to with advantage must be very fine; as, for other than a very fine model, the ship will already be beyond the part of its speed curve where the Admiralty Co-efficient holds with any approach to accuracy. All turbine ships built so far, we believe, have conformed to this; thus, the dimensions in Table III give for the "Eden" a co-efficient of fineness for displacement of .417. Even with this small value, the analysis of her results, at about twenty-five knots, given above, seems to show that the lower limit of advantage in economy over the reciprocating engine was very well over twenty knots—if not over twenty-five knots.

Passing now to speeds below the lower limit we must also conclude, from the above discussion, that the disadvantage quickly increases, and a new factor soon presents itself. We have supposed that the pressure per square foot of the propeller was kept at the maximum value for which cavitation would not occur. But as we lower the speed the diameter of propeller cannot be indefinitely diminished. A very small propeller would lose its efficiency through being screened behind the hull, and the brackets or stern post supporting the stern shaft. Also, a slow ship—say a ten or fourteen knot ship—could not keep steerage-way against strong head winds and seas, and would inevitably be driven on a lea shore, unless its propeller was kept large, and consequently its revolutions moderate.

We see no ground for the statement given in the last sentence of the foot note on page 43, and join issue with Mr. Parsons as to its accuracy.



RIEDLER-STUMPF TURBINE AND GENERATOR

VISIT TO MANCHESTER

On our visit to the Westinghouse Works at Manchester we were exceedingly interested by the turbine work in progress and delighted by the courtesy shown in giving us very full information on all questions connected with these turbines. As this is all part of the records of the Westinghouse Company it would be entirely out of place to report on it here.

RIEDLER-STUMPF TURBINE

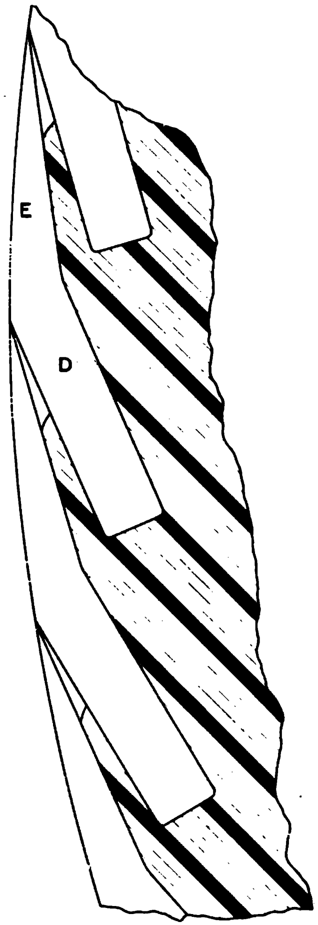
Twice, while we were in Berlin, we called at the Allgemeine Electricitäts Gesellschaft and were most courteously received. This company manufacture the Riedler-Stumpf turbine and have purchased the patent rights, for the continent of Europe, of the Curtis turbine. More recently they have bought an interest in the Parsons turbine. This seemed to convince Mr. Spiecker, of the competing firms of Seimens and Halske and Seimens-Schuekertwerke (who understood that the Riedler-Stumpf was the best of the three), that the Allgemeine Company were not satisfied with the Curtis turbine. More probably it was due to a desire to control the whole market, if possible.

On our first visit we were received by Prof. Dr. Phil. Klingenberg, one of the directors, who fully described the turbine, showing us many blue prints. He said the speed of revolution was about the same as the Parsons, with a good deal better economy. But, from other sources, we are informed that to gain good economy the peripheral velocity has to be very high. He also said that from their own tests of the Curtis it was of about the same economy as the Riedler-Stumpf, but this not only disagrees with Mr. Evers' figures given below, but also with the supposed inferiority of the Curtis to the Parsons.

Next day we were called on by Mr. Evers, Chief Engineer in charge of the Marine Department of the Allgemeine Company. He took us first to their shops in the Brunnen Strasse, then to the power station at Moabit, where one of the Riedler-Stumpf turbines is installed, and later to the Allgemeine Union Works.

They are fitting the latter works, at great expense, to manufacture turbines alone and on a large scale. For some time they have been manufacturing the Riedler-Stumpf, but will now make the Curtis there also.

The moving parts of the Riedler-Stumpf turbine are a center shaft carrying one or more nickel-steel wheels (15 per cent. nickel, 40 per cent. elongation). Each wheel is enclosed in a cast-iron casing. Much expansion having been effected in the nozzles in all cases, these casings need not be very strong. The nozzles are made out of solid rods of nickel steel, machined down and bored out.



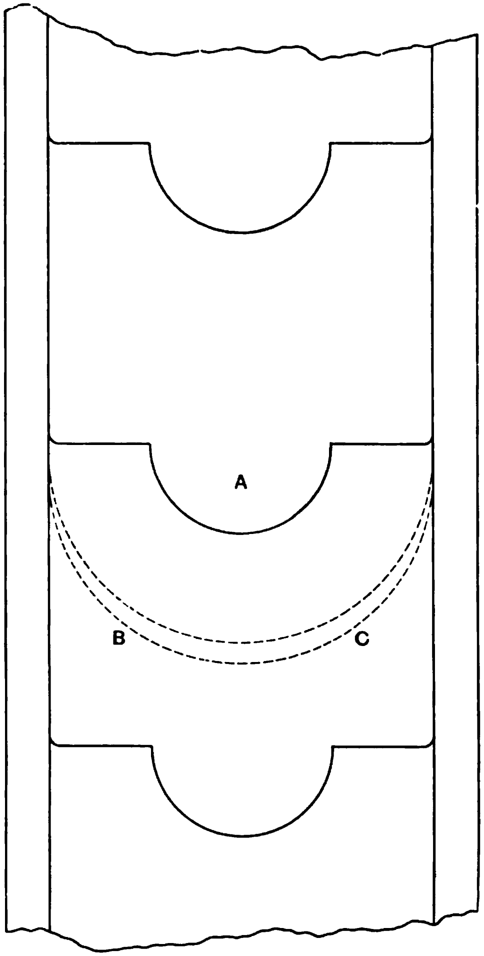


Fig. 1

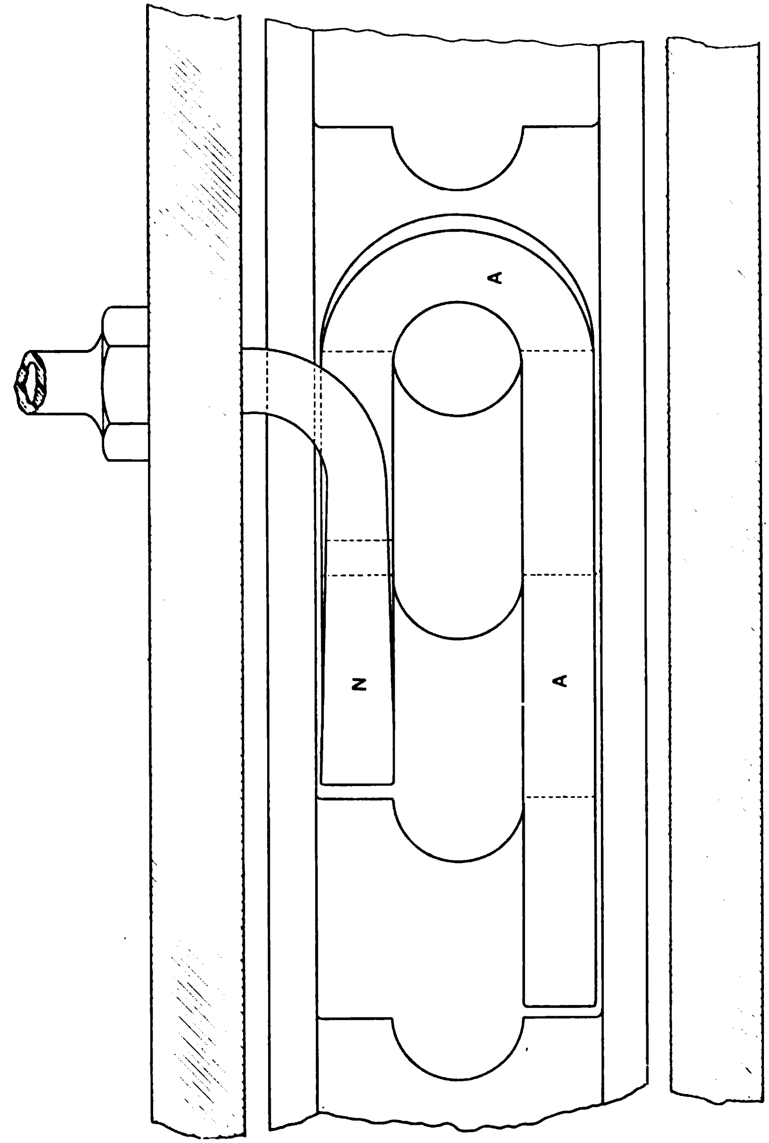
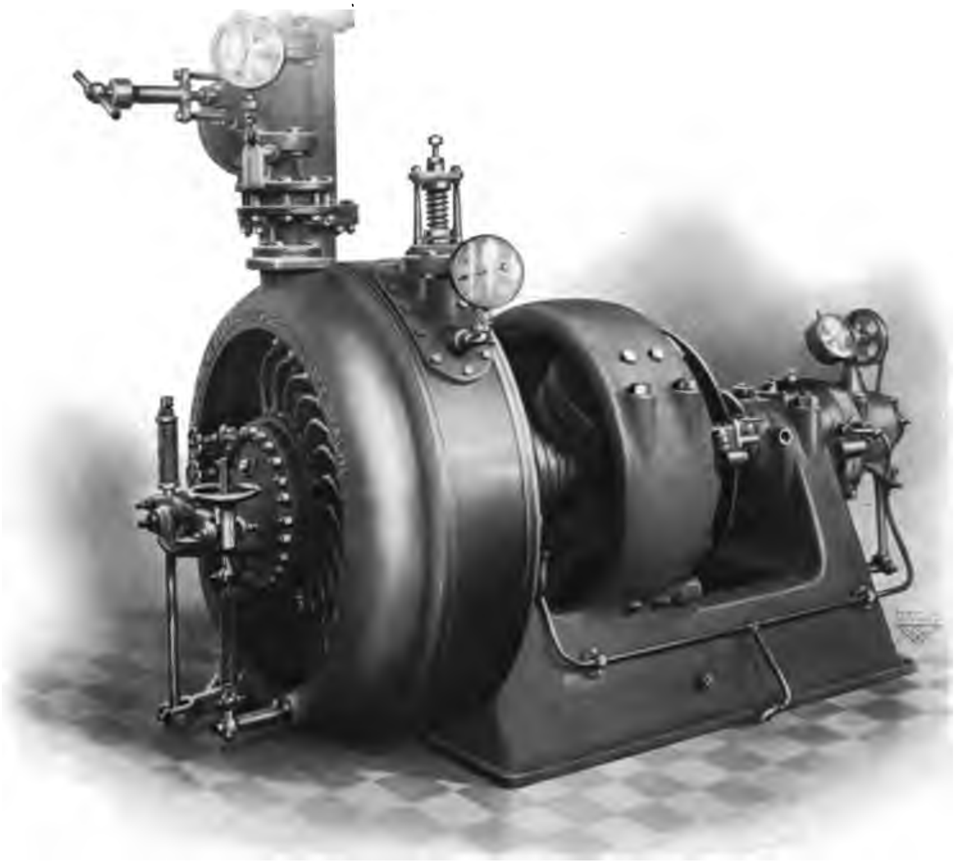


Fig. 2



RIEDLER-STUMPF TURBINE AND GENERATOR

After this they are bent round, for they enter the casing at the side and then have to turn so as to deliver the jet on the cylindrical part of the rim of the wheel. At their outlet they are also squeezed to a rectangular shape—this shape being specially patented—to give a better delivery of steam on to the wheel. The clearance from the nozzle mouth to the wheel is only two or three millimeters. The nozzles playing on the circumference of the wheel, there is nothing to preclude a small amount of end play of the shaft being allowed. This would be a considerable advantage in marine applications; which, however, we think, for reasons stated below, are not probable with this turbine.

The accompanying sketches, Figs. 1 and 2, show the shape of the buckets on the wheel, the buckets being milled out. These figures are merely diagrammatic and are not taken from working drawings.

The machine, if only one stage is used, is seen to be a modified de Laval turbine, the principal difference being in the way the buckets are formed. But in the more economical, and expensive, machines more than one stage—up to four—is used; also the steam is, in some cases, delivered twice on to the same teeth, by a bent nickel steel tube of rectangular section, as shown at A, Fig. 2. In the most economical turbines there is one wheel in a separate case for each stage, steam only being delivered once on each wheel; but for cheapness, two sets of buckets, for two stages, are sometimes cut on the same wheel, delivery from the earlier to the later stage being by a bent tube similar to A, Fig. 2.

Fig. 3 gives approximate sizes for one wheel we saw, which has a high and a low pressure stage cut on its periphery. All sizes were taken by eye except the numbers of teeth.

The only parts under very heavy stress are the wheels, and this is why the very costly material used is required. The periphery is made comparatively light, and the central web or disc is made tapering in thickness from very thin where it joins the periphery to much thicker at the hub, this shape being adopted for equal strength. The speed required for economy seems to leave a very small factor of safety in these wheels. Mr. Mentz, at the Vulcan Works, Stettin, told us that Prof. Riedler claimed a factor of safety of four, but that even this small value was disputed by others. This high peripheral speed and the very low factor of safety of the wheel has evidently been the subject of very general remark, as they were also mentioned by Mr. Zoelly, at Zurich, and Mr. James Sulzer, of Messrs. Sulzer Bros., Winterthur. It seems to be an element of considerable danger

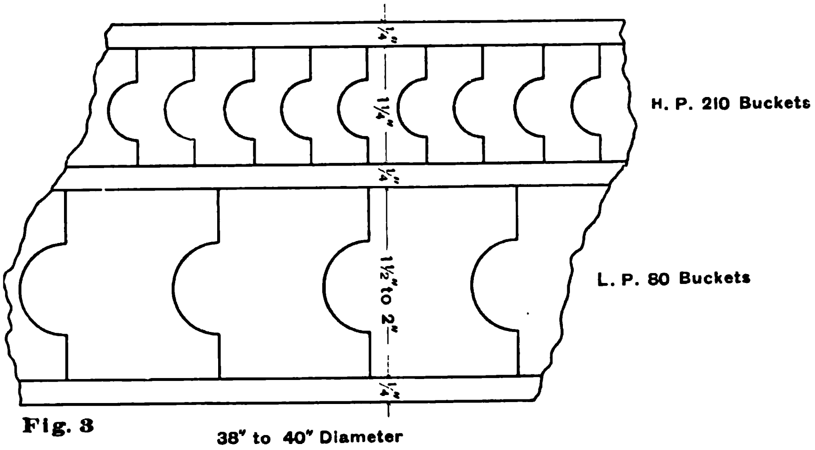
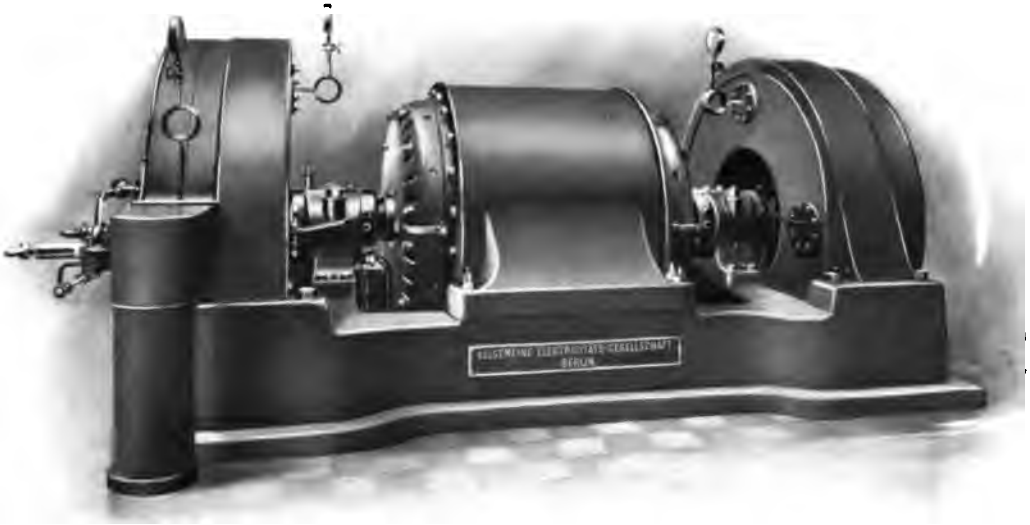


Fig. 3



RIEDLER-STUMPF TURBINE COUPLED WITH 3-PHASE ALTERNATOR

in the design, and Mr. Sulzer said that one disc had been found cracked, but the discovery was made in time to prevent disaster. If it had been running with the crack it rather seems to prove that the factor was higher than the above estimates. Without knowing the exact circumstances it is impossible to judge.

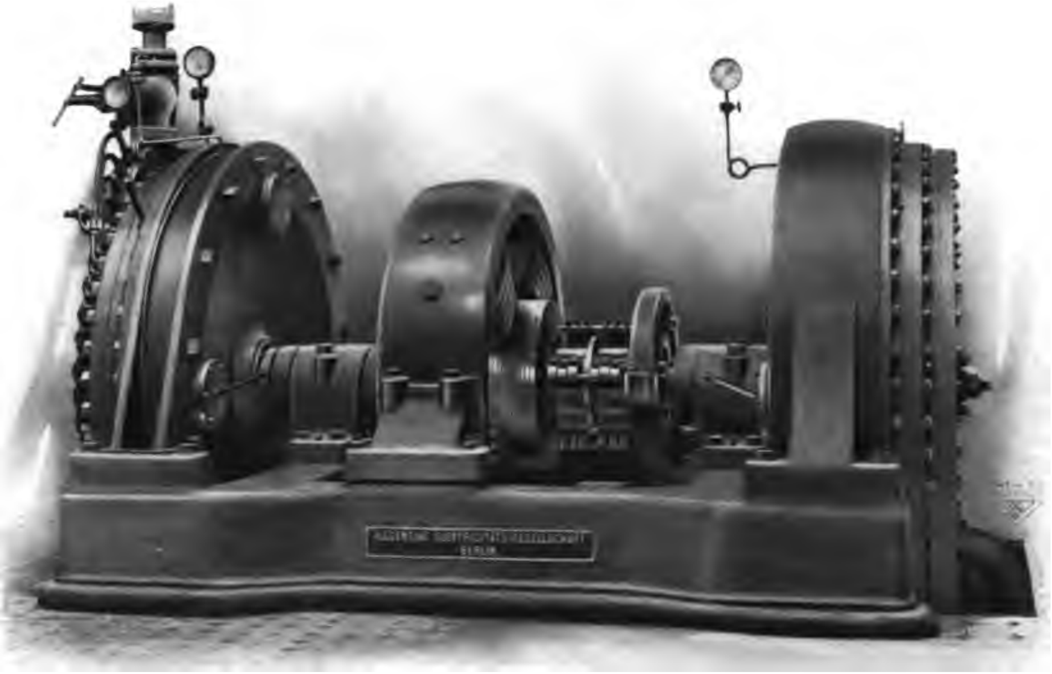
The workmanship was of an exceedingly high class; the steam passages through the nozzles were polished as bright as a gun barrel, and the wheels finished with the greatest nicety. It is a point on which they laid the greatest stress, and which, they said, experiment had shown to be a necessity if good economy was to be attained. We can readily credit this statement as in this way alone could vortex motion be minimized.

Their latest machine for milling the buckets consisted of a large horizontal table to which nine small milling machines were attached. Each of these machines was driven by a separate motor. The turbine wheel was clamped, with its axis vertical, to this table, and all nine machines operated at once. The work presented a beautifully fine appearance when finished and the edges of the buckets presented to the steam were very thin.

A center hole at A, Fig. 1, was first bored, then the millers inserted to work out B and C, which were said to be of different radius to accommodate the expanding steam. The surfaces at D and E were also at a slight angle, as shown.

At first the high-pressure nozzles were equally distributed round the circumference of the wheel; but now they are all placed as close together as possible so as to deliver only on to one portion of the circumference, extending to from one-fifth to one-quarter of the whole. Tests showed that this increased the economy from twelve to fourteen per cent.

The control of this turbine is through a governor at the high-pressure end. If the revolutions increase above those intended, a spring band is loosened so as to cover up the ends of one or more of the pipes leading to the high-pressure nozzles. Thus there is little wiredrawing.



RIEDLER-STUMPF TURBINE AND GENERATOR

This turbine has a very small number of parts, especially running parts, compared with any of the types now in the field—compared even with the de Laval. Although the manufacture of the nozzles and wheels must be costly there is not a very great deal of this work, and we do not think the total cost of manufacture should be very high.

Indeed, the two features which struck us most forcibly were the simplicity and shortness of the machine. Even if made considerably larger in diameter than a corresponding Parsons turbine, to gain a high peripheral velocity, it may still prove to be a good deal lighter.

The drawback of the turbine is that, like the de Laval, which it most resembles, it requires a very high peripheral velocity for the best economy. Mr. Evers gave this as 340 meters (= 1,115 feet) per second, and he said the turbine falls very rapidly in efficiency if this speed is reduced. Thus, in a very early pattern, which we saw laid aside at the works, a disc of two meters diameter was run at 750 revolutions per minute, giving a peripheral speed of 78.54 meters (= 258 feet) per second, driving a four-pole generator; but the efficiency was found to be exceedingly low. The generator will now be made two-pole and the turbine driven 1,500 revolutions per minute (= 515 feet per second peripheral speed) from which a much better, but still not very good, efficiency is expected. But with the high peripheral speed of 1,100 feet per second they claim, and we have little doubt they will realize, a good economy.

The Allgemeine Company having arranged to build the Curtis turbine in Germany they imported a 6,000 K. W. turbine from America. The following results comparing the Riedler-Stumpf and Curtis turbines will be of interest; but Mr. Evers said they considered the workmanship of this Curtis turbine as poor and the pattern was old. They think that with the same fine finish as they are putting on their turbine the results will be better.

The experiments were made by the Allgemeine Company themselves and the conditions for both turbines were the same, the power being 6,000 K. W.

Riedler-Stumpf,	8.1	Kilogrammes	per K. W. hour.
Curtis,	8.6	do.	do.

The General Electric Company have arranged to manufacture the Riedler-Stumpf turbine in the United States.

The Allgemeine Company claim that for small sizes the Riedler-Stumpf is superior to all others, while for large sizes they expect it will come out about the same as the Parsons and Curtis.

The following figures were given in the course of conversation; their exact accuracy we had no means of verifying,—

Mr. Rathenau, General Director of the company, said that the total weight of a 30-horsepower Riedler-Stumpf turbine with dynamo complete was one ton.

As showing how this turbine has been reduced in weight as it has developed, Mr. Evers gave the following figures,—

Old pattern, 75 K. W., 7,000 kg., 30 lbs. water per K. W. hour, one stage;

New patern, *100 K. W., 4,500 kg., 21.5 lbs. water (expected) per K. W. hour, 3,000 revolutions per minute, 4 stage, 210 pounds steam pressure, 200° superheat.**

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* This refers to a machine we saw building. The anticipated consumption of water was founded on the results from previous turbines.

** We think the Centigrade scale was meant, but we are not quite sure.

Evidently the Allgemeine Company believe thoroughly in the future of the turbine and in the turbines they are manufacturing especially, as they are sinking much capital for its manufacture. Indeed, Mr. Rathenau's solicitude and commiseration for the builders of reciprocating engines seemed to us not a little premature.

The future of the Riedler-Stumpf turbine on land may be considerable if it is found safe to run, and if the relative economy is as good as stated by the Allgemeine Company; but on this last point there seems to be considerable doubt in Germany and it was the subject of very acrimonious dispute in the discussion of a recent paper by Prof. Riedler before one of the German technical societies. The dispute was principally with the advocates of the Parsons turbine and happened prior to the recent combination. Mr. Mentz felt that considerable doubt had been cast on the economy as stated by Prof. Riedler; while Prof. Flamm thought that probably the economy of both was nearly the same, the diameter and peripheral speed of the Riedler-Stumpf being, however, both somewhat greater than the Parsons.

We think that for marine purposes there is little future for this turbine, the high peripheral velocity completely blocking the way. This requires either very high speeds of revolution or very large diameter, both of which are inadmissible. And this conclusion seems to be generally held. Indeed, Mr. Evers himself said that they had had an inquiry for a marine application, but the Company would not consider it as they could not procure economy of the turbine at a reasonable speed of revolution.

ZOELLY TURBINE

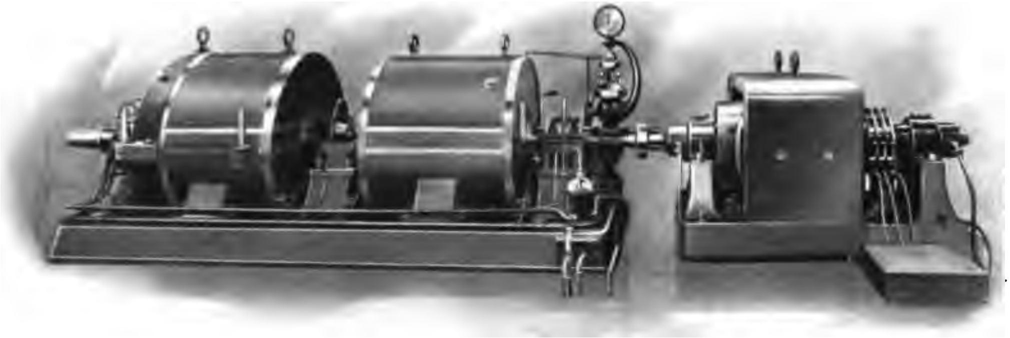
We visited the works of Messrs. Escher Wyss & Co., Zurich, and saw Mr. Zoelly, who seemed anxious to give us all information regarding his turbine. This is a turbine of which Mr. Spiecker of Seimens and Halske spoke, in which his firm, together with Krupp, Creusot, Nurnberg, and the North German Lloyds, and other firms, are interested.

It is an "action" or "impulse" turbine and in its most recent type purely parallel flow—as shown in the right-hand portion of Fig. 66 of Prof. Stodola's book on Steam Turbines (first German edition), and reproduced in Fig. 4 of this Report. The machine consists of alternate fixed discs and rotating steel discs mounted on a shaft. The fixed discs have circumferential rectangular nozzles cut at a comparatively small angle to the plane of rotation. The steam from these acts on the curved blades of the next moving disc, the steam passing in succession through the fixed and moving discs and the steam passages being enlarged, as usual, as the pressure falls.

The nozzles are the only passages through the fixed discs (except the slight clearance at the shaft), hence the leakage is little through these. Inside the chambers the leakage past each wheel is also very small for reasons given in the next paragraph.

In "action" turbines all the expansion is effected in the nozzles; consequently the pressure in each chamber is practically uniform, having the same value on each side of the running wheel. This gives two great advantages over the "reaction" turbine,——

1st. The tendency to leakage round the ends of the running wheel is very slight, and is further reduced by placing the nozzles a little way in from the ends of the vanes. Hence between the ends of the vanes and the case may be as large as 2 to 3 millimeters (.08 to .12 inch) and there is no danger of the blades being stripped through contact with the casing arising from unequal expansion due to heat or through wear of the bearings.



ZOELLY TURBINE AND ELECTRO-MOTOR

2d. Axial pressure does not exist and there is no necessity for balancing pistons. This is a great constructional advantage, being not only a simplification but doing away with a source of leakage and danger.

Mr. Zoelly also stated that without sensible loss of efficiency the fore and aft clearance between the nozzles and the rotating vanes could be made as large as 5 or even 10 millimeters (.2 to .4 inch). If this holds, it would allow, in marine applications, of the turbine and thrust shafts being bolted up solidly without any fear of accident from the wear of the thrust block.

The construction of the turbine is given in Figs. 4 and 5, which are taken from Prof. Stodola's book. The center parts, A, of the moving discs are of ordinary steel, the blades of nickel steel.

The nozzles are not flaring as in de Laval's turbine, but parallel. Founding on a well-known result regarding the flow of steam through nozzles, Mr. Zoelly states that if the change of pressure in one step (from p′ down to p) is such that

$$\frac{p'}{p} < 1.73, \text{ or } \frac{p}{p'} > .58,$$

the parallel nozzle may be used. Mr. Zoelly uses p′/p = 1.64 and claims his results bear out the theoretical indications. This value reduces the pressure from 10 atmospheres absolute to about one, in five steps, in which case the turbine would be divided here and an outside bearing interposed. The stuffing boxes on each side of the bearing have then nothing to do as they have atmospheric pressure on each side.

The revolving discs have been shown to be amply strong. That is, a disc

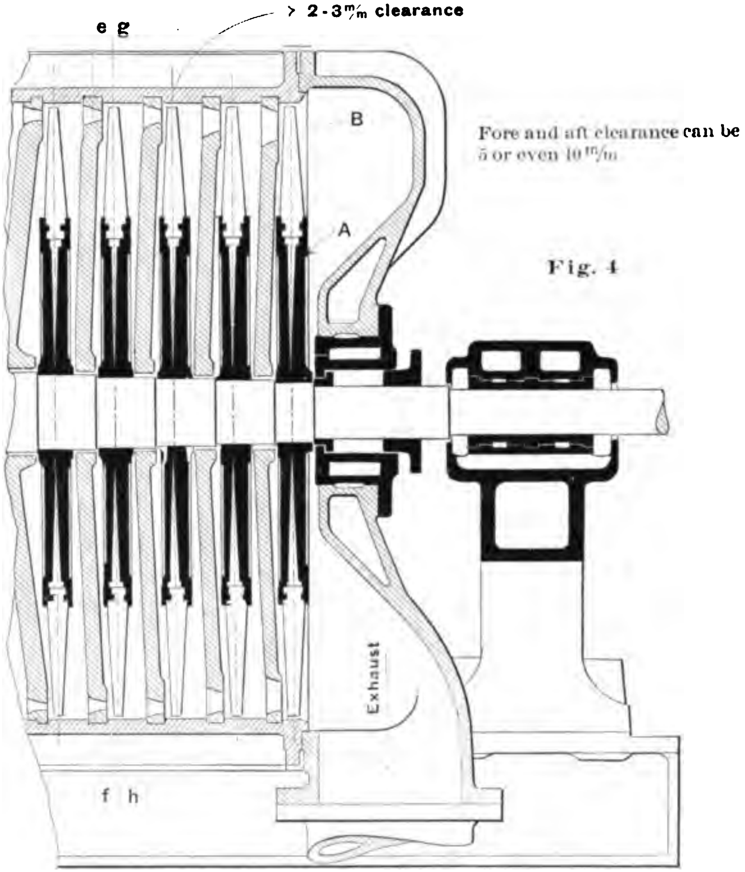
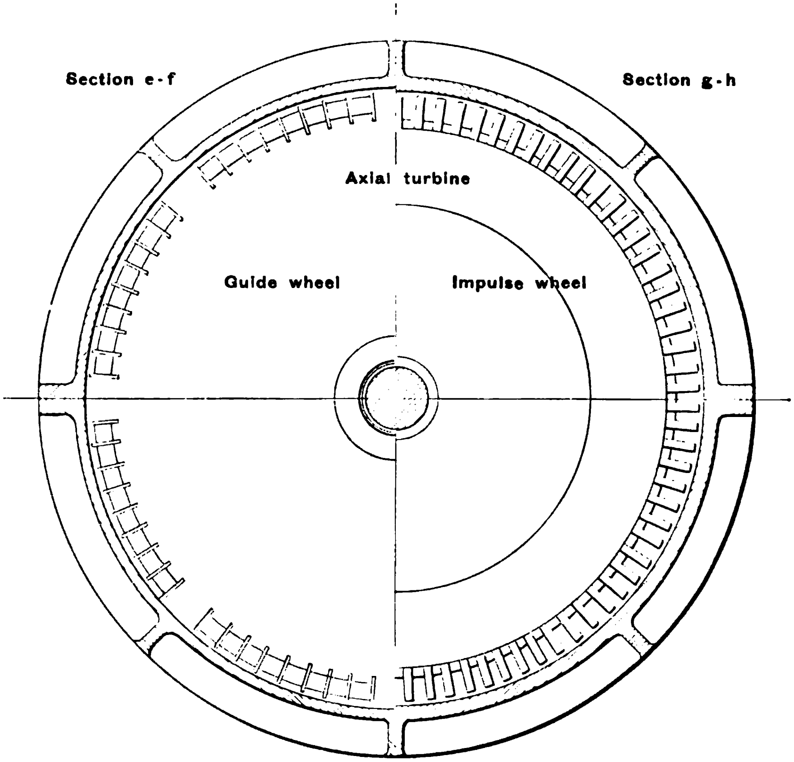


Fig. 4

Fig. 5



designed to run at 3,000 revolutions per minute has been experimentally run at 7,500 revolutions per minute with no failure.

It is proposed to make the smaller turbines with 10 moving discs, increasing to 14 in the large turbines, the increase being made to diminish the peripheral velocity. This increase is necessary if high efficiency is to be maintained, since the proportionate fall of pressure has also to be diminished.

The proposed peripheral velocity for ordinary purposes is 150 meters per second; but by the increase in the number of discs to not more than 14 it is *proposed* to reduce this to 70 meters or even less per second for large marine turbines. Mr. Zoelly states that with discs not exceeding 2½ meters diameter he can run these turbines at 350 revolutions per minute.

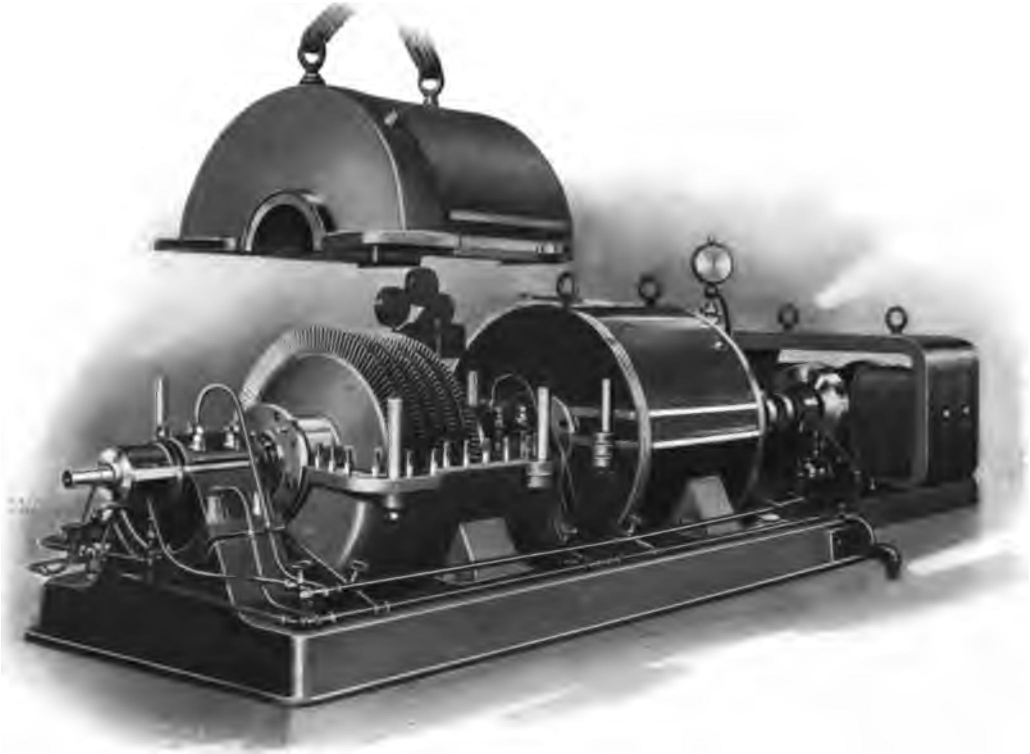
$$\text{Circ } 2\tfrac{1}{2} = 7.854$$

$$\frac{350 \times 7.854}{60} = 45.8 \text{ meters per second.}$$

This, of course, would require more than 14 steps.

He explained that an advantage of the "action" over the "reaction" turbine lay in the fact that, if v is the steam velocity, the latter requires a peripheral velocity for high efficiency of .65 v while the action turbine may have as low as .5 v or even .37 v, and therefore for the same peripheral velocity may have larger drops of pressure giving much fewer steps. Consequently a 10 step Zoelly turbine may be made with, say, 2,000 pieces, while the corresponding Parsons turbine would have 30,000.

Since the general character of the efficiency curve is as shown in Fig. 6, while the maximum efficiency is at .5 v, not much loss is suffered by adopting .35 v. By adopting a higher velocity, as at C, the efficiency may be made to rise as the peripheral speed is reduced; as, for instance, when slowing the speed of a ship driven by a turbine. This, however, would mean a very high peripheral velocity at full speed, which is not admissible on board ship.



ZOELLY TURBINE WITH COVER LIFTED

A WHEEL OF THE ZOELLY TURBINE

The reasons why large turbines (see Table VII) are more efficient than small are,—1st. In the small guide discs the nozzles go around only part of the circumference and are very small radially, while in large machines they go all around and are larger. 2d. In the small design the point B of the above efficiency curve is aimed at, for cheapness, while in the large design, A is attained. By extra cost A could be attained for all.

We saw one 600 h. p. turbine running very satisfactorily at 3,000 revolutions per minute. Probably this is the turbine tested by Prof. Stodola, the results of which we give in Table VI.

TABLE VI.

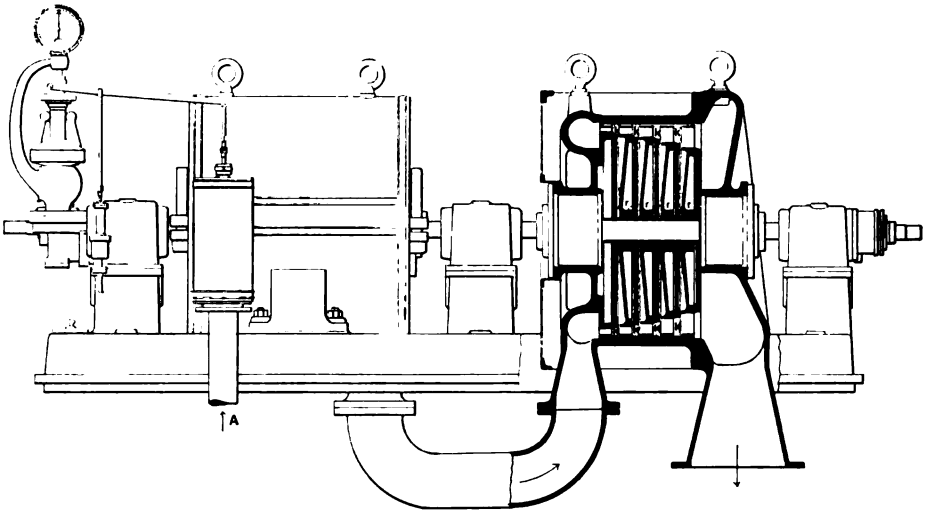
TEST OF ZOELLY TURBINE BY PROF. STODOLA

Boiler pressure = 10.5 Atm. = 154 lbs. per sq. in.

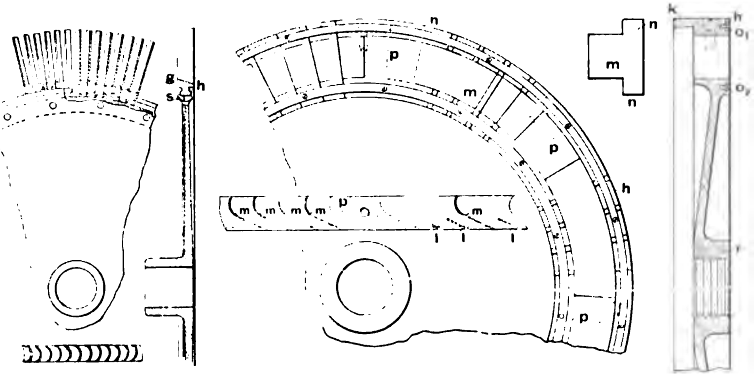
Vacuum about 93.5 per cent.—say 28 inches.

The power was measured in kilowatts and the weights of steam in kilogrammes. The figures for horse-power and pounds weight were calculated from these by us.

I. SATURATED STEAM.	Running Empty without Exciter.			Steam per Hour 295.4 Kgr.=651 Lbs.		
	Running Empty with Exciter.			Steam per Hour 465.4 Kgr.=1026 Lbs.		
	KILOWATTS.	HORSE POWER.	TEMPERATURE OF STEAM, FAHR.	STEAM PER K. W. PER HOUR. KILOGRAMMES.	STEAM PER K. W. PER HOUR. POUNDS.	STEAM PER H. P. PER HOUR. POUNDS.
	80.1	107	368°	15.0	33.0	24.67
	182.2	244	368°	11.7	25.8	19.25
	240.1	322	368°	10.9	24.0	17.9
	334.5	448	368°	10.1	22.27	16.6
	387.6	520	368°	9.74	21.48	16.0
II. SUPERHEATED STEAM.	390.4	523	428° 60° super-heat.	8.98	19.8	14.8
	391.7	525	464° 96° super-heat.	8.63	19.0	14.2



ZOELLY TURBINE—PARTLY IN SECTION



DETAILS OF ZOELLY TURBINE

TABLE VII.

STEAM CONSUMPTION OF ZOELLY TURBINES.

Vacuum 92%=say, 27.6 inches.

Size of the Machine.	400 K. W.						1,000 K. W.						3,000 K. W.						5,000 K. W.					
Admission Pressure. (Steam).	10 Atm.= 147 Lbs.			13 Atm.= 191 Lbs.			10 Atm.= 147 Lbs.			13 Atm.= 191 Lbs.			10 Atm.= 147 Lbs.			13 Atm.= 191 Lbs.			10 Atm.= 147 Lbs.			13 Atm.= 191 Lbs.		
Condition of Steam as Regards Temperature.	Saturated.	250° C.	300° C.	Saturated.	250° C.	300° C.	Saturated.	250° C.	300° C.	Saturated.	250° C.	300° C.	Saturated.	250° C.	300° C.	Saturated.	250° C.	300° C.	Saturated.	250° C.	300° C.	Saturated.	250° C.	300° C.
Steam Consumption in Kgr. per K. W. Hour. Kilowatts Measured at Terminals.	10.3	9.7	9.3	9.9	9.3	9.0	9.4	8.9	8.6	9.1	8.6	8.3	8.5	7.9	7.65	8.1	7.6	7.2	8.2	7.6	7.3	7.8	7.3	7.0
Kgr. per Brake H. P. Hour.	7.0	6.55	6.3	6.7	6.3	6.1	6.6	6.2	6.0	6.4	6.0	5.8	6.0	5.6	5.4	5.7	5.35	5.1	5.8	5.35	5.15	5.5	5.15	4.9

250° C.=482° F., which is a superheat of 118° F. for steam at 147 lbs. pressure per square inch.
250° C.=482° F., which is a superheat of 104° F. for steam at 191 lbs. pressure per square inch.
300° C.=572° F., which is a superheat of 208° F. for steam at 147 lbs. pressure per square inch.
300° C.=572° F., which is a superheat of 194° F. for steam at 191 lbs. pressure per square inch.

Inclusive condensation 3%. (The meaning of this expression is not quite clear, but we think it may mean that 3% is to be added to the above values if the power required for condensation — driving the air and circulating pumps — is supplied by the turbine).

Table VII gives a series of results for different sizes of turbines. These results have been deduced mostly from an extensive series of experiments on single units; that is, on single moving discs, not from complete turbines.

The best result is,—

5,000 K. W. turbine. 300° C. temperature of steam. Steam pressure, 13 atmospheres. Vacuum, 92 per cent. Steam consumption, 4.9 kgr. per brake horse-power hour.

From this Mr. Zoelly deducted 10 per cent. for improved results of more recent experiments, and another 10 per cent for mechanical efficiency; i. e.,

$$\frac{\text{B.H.P.}}{\text{I.H.P.}} = 9$$

$$\begin{array}{lr} & 4.9 \\ \text{10 per cent. off} & \underline{.49} \\ & 4.4 \\ \text{10 per cent. off} & \underline{.4} \\ & 4.0 \end{array}$$

The Zoelly steam turbine is governed in the same manner as the hydraulic turbines of the same firm, viz., "by an exceedingly sensitive spring-regulator with Serv-motor, whereby an initial pressure of the steam is changed correspondingly to the momentary load." We did not see this apparatus.

Mr. Zoelly has very large experience with hydraulic turbines and has also made extensive experiments with steam turbines for four years. We were very favorably impressed by his evidently great knowledge of the whole subject. Mr. James Sulzer, who knew Mr. Zoelly well, spoke very highly of his ability and thought well of the turbine, though he also believed there was still need of further experiment. Messrs. Escher Wyss & Co. think, on the other hand, that their preliminary

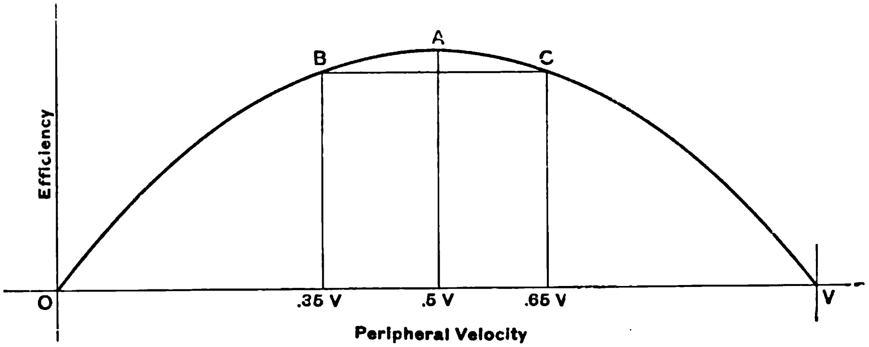
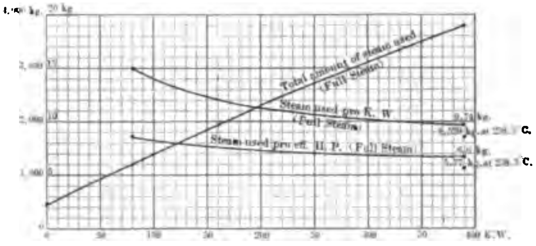


Fig. 6

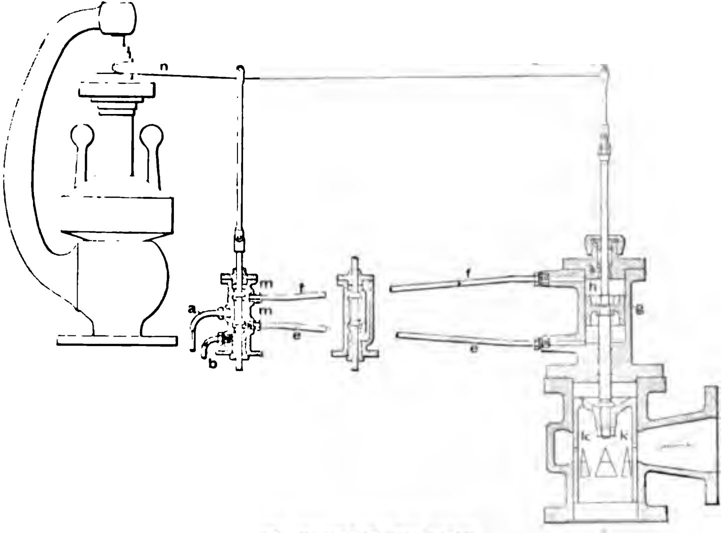
experiments may be looked on as complete and, when we were there, were about to call in other eminent experts besides Prof. Stodola to give certified results of performance.

Mr. Sulzer said Messrs. Brown, Boveri & Co. had run tests some time before and had found the economy as good as with the Parsons turbine. The builders believe it is better, and Mr. Spiecker said they had convinced the North German Lloyds Steamship Company that it was superior to the Parsons, Riedler-Stumpf, or Curtis, and these steamship owners have purchased an interest in it.



TEST OF ZOELLY TURBINE

We strongly believe that it would be well to test the merits of this turbine.

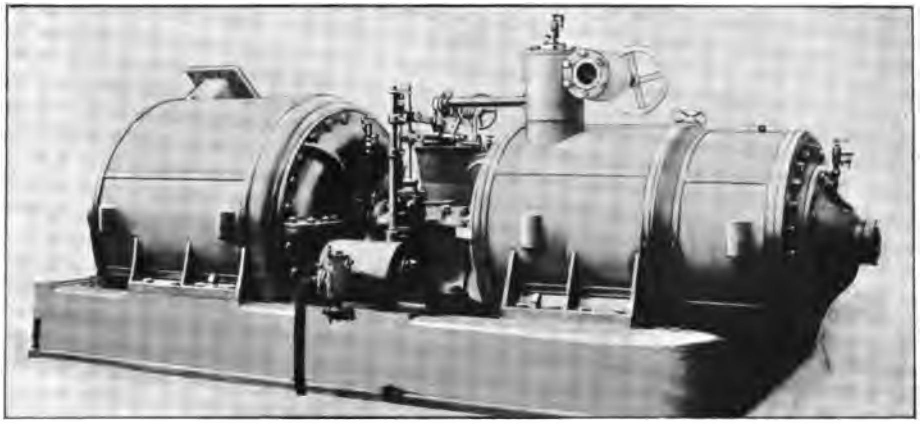


REGULATING DEVICE—ZOELLY TURBINE

RATEAU TURBINE

We visited the works of Messrs. Sauter Harlé & Co., Avenue Sufrein, Paris, who have been the principal makers of the Rateau turbine. We were courteously received by the Superintendent of the Works, who discussed turbines freely and also showed us the work under construction. There have been three sets of Rateau turbines built for ships,—

1st. One set of 2,000 h. p. for a French torpedo boat, with two shafts. The revolutions were from 1,700 to 1,800 per minute.

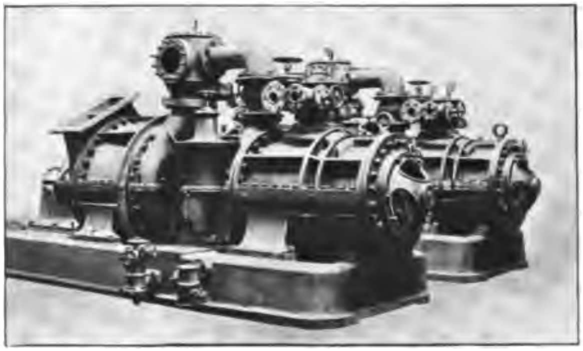


RATEAU TURBINE AND GENERATOR

2d. One set, of about 2,000 h. p. for Messrs. Yarrow & Co., to which we have referred already. These turbines were made at the Oerlikon Works in Switzerland.

3d. A set for the French Government which had not been installed when we were in Paris.

The Superintendent said they had not been successful with their marine applications. The first he seemed to look on as a complete failure, arising from the very fast revolutions of the shafts and the consequent poor performance of the propellers. He said they know now the revolutions must be much slower, and he also spoke of the large consumption at low speeds. With Messrs. Yarrow & Co.'s boat he said they had done a good deal better, but he evidently did not think it a success. Though he said the speeds of revolution had been considerably reduced here from those of the first application, we find from Prof. Rateau's paper of 25th March, 1904, read before the Institution of Naval Architects in London, that in one trial one shaft ran as high as 1,657 revolutions per minute; and, in the last trial reported by Prof. Rateau, at 1,307 revolutions per minute. The badness of the results of both of these installations is fully confirmed by Prof. Rateau's paper. This paper of Prof. Rateau's is a very instructive one to read in connection with this Report. Mr. Martin of Flushing also remarked to us that Mr. Yar-



RATEAU TURBINE AND PUMPS

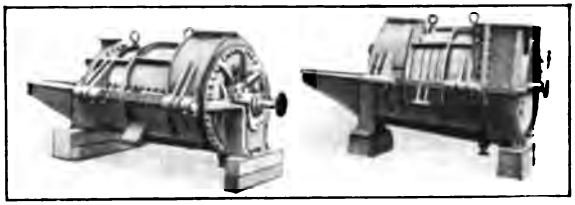
RATEAU TURBINE

row had not done very well with his turbine boats; though Mr. Yarrow himself told us he had done better with the Rateau than with the Parsons turbine.

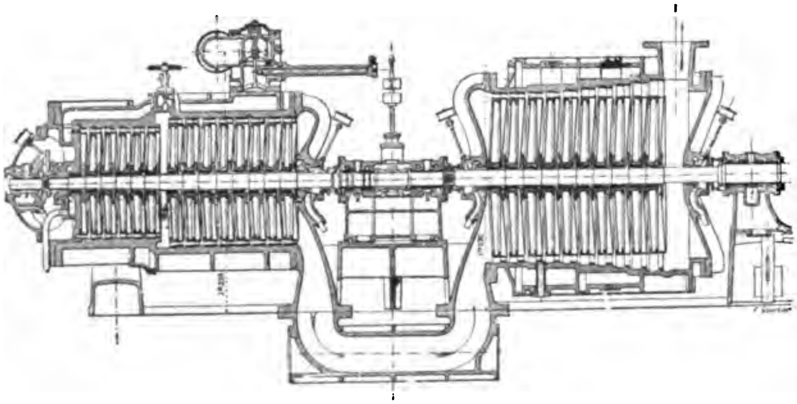
The Rateau and Zoelly turbines belong to exactly the same class, but their construction is very different.

In the Rateau, the fixed diaphragms, with the guide blades or nozzles, are made of cast steel or cast iron as the strength required demands.

The moving discs are built up. A circular steel plate is flanged at the central hole and rim and is mounted on the shaft. In one machine which we saw open this disc was about $\frac{1}{4}$ inch thick and had lightening holes. The blades are made of steel pressed to the desired curve and ground to sharp edges where the steam strikes and leaves them. A flange is turned on the blade at one end and by this it is fastened with a single steel rivet to the outer rim of the flanged plate just mentioned. At the outer end of the blade a projection about $\frac{3}{16}$ or $\frac{1}{4}$ inch broad was left which passed through a corresponding rectangular hole in an outer circumferential ribbon and was riveted over. This ribbon was very thin and was not all in one piece. The balancing of each disc was carefully carried out on a specially constructed and very delicate machine.



RATEAU TURBINES



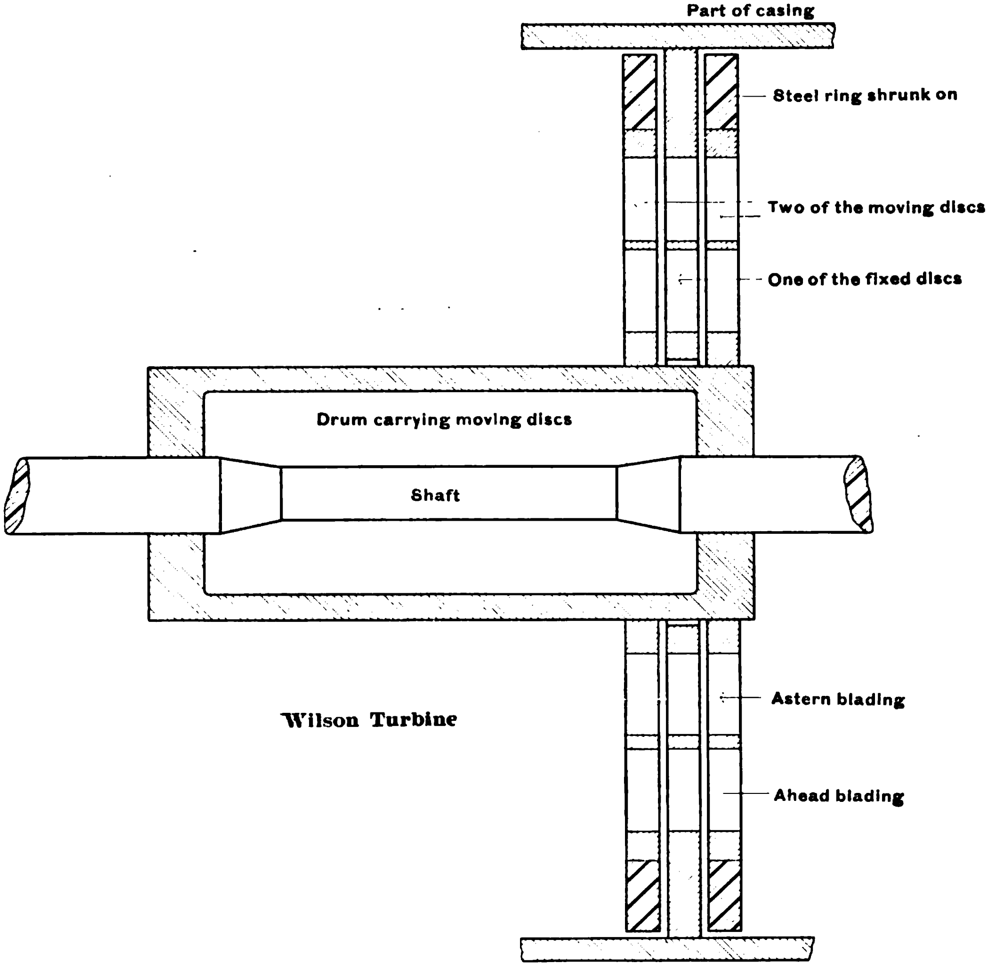
SECTION OF RATEAU TURBINE

We agreed that the whole construction seemed cheap; so much so that we doubted if the work was good enough to be durable.

WILSON TURBINE

Mr. George Wilson, now at Greenock, Scotland, showed drawings of his direct-combustion and steam turbines, which, in the actual turbine part, are much the same.

They are parallel, double flow, and some of the designs are reversing. The gas or steam passes through a series of fixed and moving blades made of cast iron. Each disc is cast in fan-shaped pieces, each piece containing a number of radial blades, a circumferential section being much the same as looking on the ends of Parsons blading. The castings, of which we saw samples, are exceedingly rough and would be most difficult to make smooth. But Mr. Wilson does not propose to attempt this, as he does not consider it any advantage.



Wilson Turbine

Fig. 7

Fig. 7 gives a rough indication of the essential features of this turbine. The sections of the fixed blades are held together by being fixed into the outer casing; while those of the moving blades are bound by a steel ring shrunk on. The extra weight of these rings was considered an advantage by Mr. Wilson, as it added much to the flywheel action. The usual tongues and grooves in the faces of the fixed and moving rings, though not shown in the figure, were used to prevent radial leakage, at the outside and inside and also at the ring dividing the ahead and astern blading.

To go ahead a set of inlet valves were opened opposite the ahead blading; and, to reverse, these valves were closed and, by the same operation, others opposite the astern blading were opened.

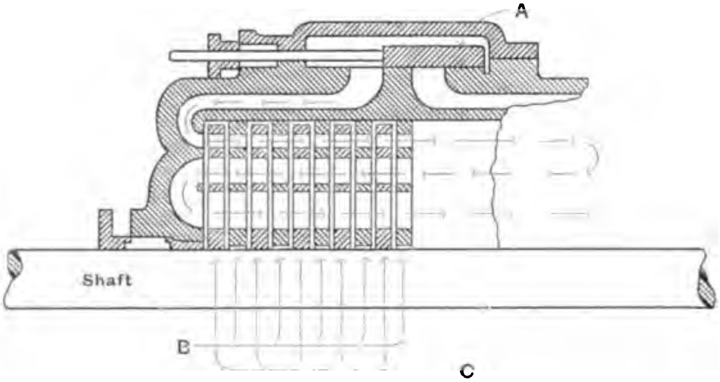
The first turbine is being built at Messrs. Scott's in Greenock, and Mr. Wilson seemed exceedingly sanguine of success.

This turbine does not seem to us to possess the elements of success, and we think there is almost sure to be much waste from viscous resistance, and also great leakage, notwithstanding the tongues and grooves, between the ahead and astern blading, so that one will act against the other. Several small points on which Mr. Wilson set great store for increased economy and consequent success seemed trivial.

He promised to send information if his trials were successful.

BUCHHOLZ TURBINE

At the rooms of Messrs. Illy & Co., Electrical and Mechanical Engineers, 8 Fitzroy Street, London, W., we met Mr. E. Buchholz, inventor of a turbine, of which he showed us drawings and a small model running. It is reversible and



Buchholz Turbine.
Fig. 8

consists of a series of fixed and movable discs B and C, Fig. 8; the former being held by the case and the latter by the shaft.

Each disc has several series of holes, Fig. 9, those in the fixed discs being bored parallel to the shaft axis, and those in the movable discs at an angle. The arrangement is somewhat like a syren, but the steam passes through all the holes of the outer row, back through the next row, and so on, as indicated by the arrows in Fig. 8.

The turbine is reversed by reversing the flow of steam by means of the valve A.

Adjacent discs, as in the Wilson turbine, are actually fitted tongue and groove between each ring of holes, Fig. 10.

The best arrangement of holes did not seem to be settled but, strangely, the inventor seemed to think a very slight angle of the holes in the movable discs was best.

The experiments for testing the efficiency have not yet been made.

We think that the losses due to fluid friction will be even larger than in the Wilson turbine, as such large surfaces run close together at a high speed; in fact, the machine would make a very powerful water brake.

Then the area for leakage round the holes would be a very considerable fraction of the area of the holes unless the discs were exceedingly close, in which case the brake action would be intensified.

As the drawings were made, the steam passed last through the inner ring of holes. So arranged it would be difficult to keep the areas increasing in proper proportion to accommodate the expanding steam, as is carefully done in all turbines which have so far given any promise. And if the areas increase in the direc-

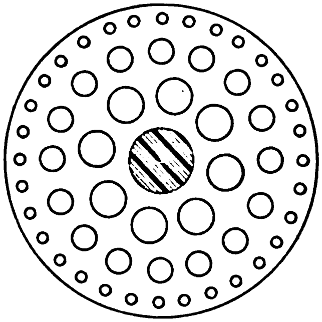


Fig. 9

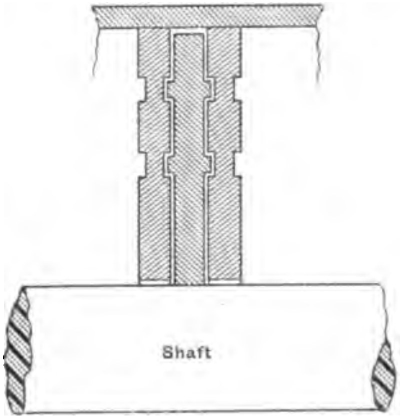


Fig. 10

tion of the flow of steam when the turbine is going ahead, it will be met by a diminishing area when going astern; leading, no doubt, to a very low efficiency.

We think this turbine will give exceedingly poor results.

VISIT TO MESSRS. BROWN, BOVERI & COMPANY

We visited the works of Messrs. Brown, Boveri & Company, at Mannheim in Germany, and Baden in Switzerland. In Mannheim they had not yet commenced turbine work.

At Baden we met Mr. Brown and Mr. Ridsdale; the latter being a naval architect who looks after all shipwork, propellers, etc., for the firm.

They were building only Parsons turbines and we saw little that apparently differed from the practice at Pittsburg. For some barrels of the turbines, however, they were using weldless steel tubes, as in the Westinghouse turbines at Manchester.

The blades near the high-pressure inlet and for a good many rows along were made with a very large proportion of copper in their composition. In some cases, they said, they used nearly pure copper.

Also, the blades seemed to us of less variable thickness than, judging from memory, were those we had seen at Pittsburg. See Fig. 11.

Mr. Ridsdale said that in some Parsons turbines they had installed at Frankfurt the consumption of water per *indicated* horse-power per hour was 9 pounds; which, he thought, broke the record. He did not know what had been taken as the estimated ratio of indicated to brake horse-power.

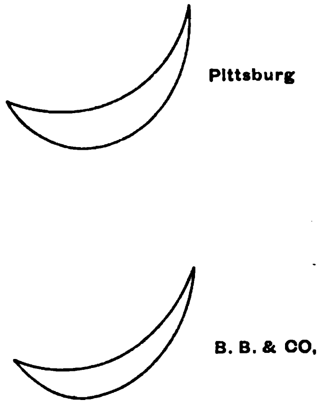


Fig. 11

AUXILIARIES FOR TURBINE SHIPS

The following information was from a large maker of auxiliaries in Great Britain.

Air Pumps.—The British Admiralty rule for independent air pumps, direct-acting, is 75 cubic inches per I.H.P. per minute. This, our informant thought too large; but Mr. Parsons increases it to 100 cubic inches per I.H.P. per minute for his turbines.

Centrifugal Pumps.—The "Eden" has two 14-inch centrifugal pumps. All the other ships of the same class and power have two 12-inch pumps.

CONCLUSION

No one can doubt that the future opens a very wide field for the steam turbine on land. In the many cases where a high speed of revolution is an advantage, the turbine is a simple and economical motor.

The limits of the field for the marine turbine are not so well defined. In the past there have been comparative failures and successes, and in the future we have no doubt there will be failures and successes also, till we gradually learn more perfectly what are the conditions of success. We have already stated that we have been led to believe it would be injudicious to apply the turbine to other than very fast ships which have to run a very small proportion of their time at cruising speed; and even in the case of fast ships the advantages have been far overstated.

If one could devise a means of reconciling, in a practical manner, the necessary high speed of revolution of the turbine with the comparatively low rate of revolution required by an efficient propeller, the problem would be solved and the turbine would practically wipe out the reciprocating engine for the propulsion of ships. The solution of this problem would be a stroke of great genius.

The Hon. Charles A. Parsons, while being far from the first worker in this field, is rightly recognized as the leader above and beyond all others. His will be known in history as one of the great names of our profession.

May, 1904.

Melville
Rear Admiral U.S. Navy
Retired

John H. Macalpine.

Melville & Macalpine

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